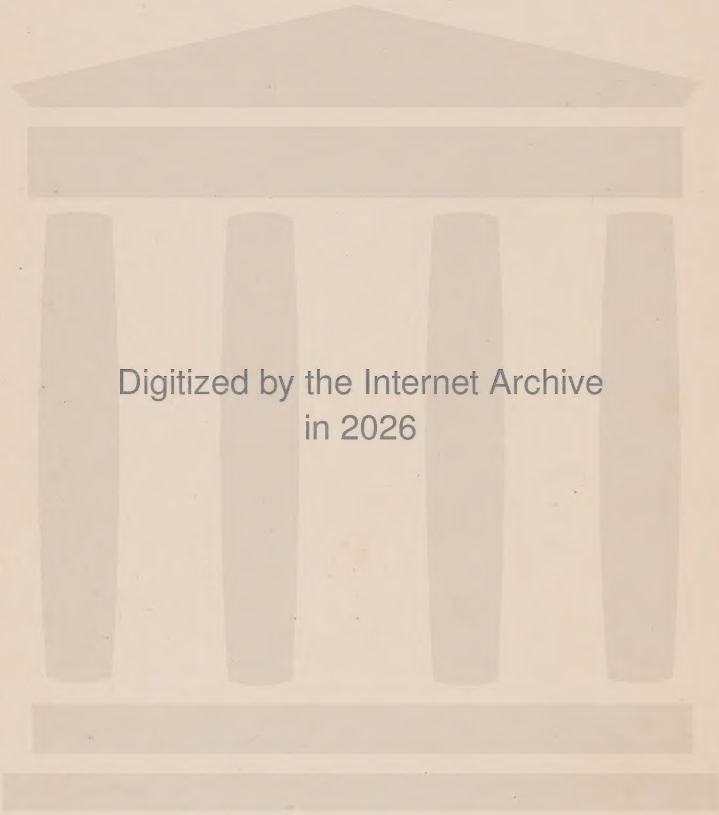
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ELEMENTS
OF
HEAT-POWER ENGINEERING

BY

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PREFACE

In preparing this textbook the Authors have attempted to include in a single volume not only the elementary thermodynamic theory of gases and vapors and of their cycles, but also the consideration of the sources of heat, the methods of making it available for useful purposes, its utilization in the various types of heat-driven prime movers and their auxiliary apparatus, together with a discussion of the fundamental theory, the ideal and actual performance and the practical considerations connected with such apparatus. The book is prepared primarily for the use of students in Mechanical Engineering in their junior and senior years, after they have completed college courses in physics, chemistry, analytical and applied mechanics and empirical machine design. The text is supposed to be supplemented by lectures, lantern slides, a study of trade catalogues and collateral reading; and, as it is prepared primarily for students who will have separate courses in mechanical laboratory practice and in the economic problems connected with heat-power engineering, but little relating to these branches is given.

A large part of the material contained in the following pages has been used during the last four years, first in pamphlet and later in book form, as a text in the junior and senior courses in Sibley College, Cornell University. It has been revised from time to time as the necessity became apparent, and now the original matter has been practically rewritten, rearranged and considerably amplified for the present book.

To add to its convenience and value as a textbook in recitation courses, all sections are numbered, the sub-paragraphs are lettered, and sample problems are given in the Appendix.

Undoubtedly errors of various kinds will be discovered, and in order that they may be eliminated it is hoped that they will be brought to the attention of the Authors, who will also welcome any other suggestions for the improvement of the book.

The Authors express grateful acknowledgment of their indebtedness to Professor A. W. Smith, Director of Sibley College, for many helpful suggestions and criticisms during the inception and progress of this work, and to Assistant Professor Ellenwood who prepared a large number of the appended problems.

They desire to extend their thanks to Professor Lionel S. Marks and to Dr. Harvey N. Davis, and their publishers, Longmans, Green & Co., for permission to use an abstract of their steam tables, and to Professor Cecil H. Peabody and John Wiley and Sons for permission to use a reduced and modified drawing based on the former's temperature-entropy chart. Thanks are also due the following members of the Sibley College instructing staff for valuable assistance: Messrs. H. M. Parmley, T. C. Ulbricht, and R. Matthews; and to F. A. Burr, formerly Assistant Professor in the College.

July 1, 1912.

PREFACE TO SECOND EDITION

SINCE the appearance of the First Edition of this book, Professor Ellenwood has published some very valuable and extensive steam charts which have a much wider field of application and greater accuracy than have any of the Mollier Charts that have appeared. The present edition includes a discussion of these important new charts and to the Appendix has been added a small two page Ellenwood chart redrawn from the original ones, covering twelve pages. The other changes in this edition consist of a number of minor corrections and a few additions.

The authors desire to express to Professor Ellenwood their appreciation of the privilege of introducing the new charts; and they desire to thank him and also Professor Matthews and Mr. C. H. Berry for many valuable suggestions.

August 1, 1915.

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INTRODUCTORY.

1. The advancement of the human race has been largely due to the fact that man has greater ability than his fellow creatures to utilize nature's resources. At first he was driven by his own weakness to seek nature's aid for protection, and he thus became familiar with her simpler laws. This knowledge grew steadily and after a time was recorded. Now the accumulated information is too great to be grasped by any individual or group and it has become necessary to specialize. One group of specialists, the scientists, continue to delve after nature's secrets in order to add to the store of human knowledge; another group, the engineers, work to make application of discovered laws to meet the needs of humanity.

The engineer must know nature's laws and must be familiar with their applications in order that he may be able to aid the race in the development and improvement of its life. One of the most important of his problems results from the fact that man's body cannot supply the power required to carry out the conceptions of his mind. To solve this problem the engineer draws on nature's store of energy.

In general, the energy of nature's store is not directly available for human uses; it must be changed in kind or quality, transmitted through space, and made available at times of demand. The engineer must provide means for effecting these results.

One of the best examples of such changes is furnished by the conversion of heat energy into the mechanical form by means of Heat Engines. Since the world demands enormous supplies of mechanical energy, this sort of conversion is of great importance, because of the fact that immense stores of easily transportable fuel are distributed over the earth near its surface. This fuel has latent heat energy, which may be easily converted into available heat energy by combustion. It is then the duty of the heat engine to convert as large a part as possible of this

available heat energy into the more desirable form of mechanical energy.

The following pages are devoted to a consideration of transformations of latent heat in fuel into available heat, and of available heat into mechanical energy, together with a study of the devices by which the transformations are effected.

The theory of these transformations is called Thermodynamics, while the whole subject, theoretical and practical, may be called Heat-Power Engineering.

HEAT–POWER ENGINEERING.

CHAPTER I.

HEAT.

2. Heat a Form of Energy. It has been shown experimentally that heat can be produced by the expenditure of other forms of energy, and that other forms of energy can be produced by the expenditure of heat. Therefore the conclusion that heat is a form of energy is justified.

All bodies that man knows possess heat energy, "associated heat"; whatever the material or state of a body may be, it is possible to obtain heat energy from it. It is not known how this heat energy is stored in matter, but it is certainly possible, and it seems probable, that it is in some way associated with the motions and relative positions of the constituent particles. Beyond this it is not necessary to generalize here in the present state of knowledge.

3. Unit of Heat Energy. The unit of measurement of energy is based upon some effect produced by the kind of energy to be measured. Under certain conditions a rise in temperature of a body is one of the most obvious phenomena connected with an increase of associated heat; and, as the extent of this effect may be measured, it is used as the basis of the unit of heat energy.

In English-speaking countries the unit of heat energy is known as the **British Thermal Unit** (B.t.u.) and is defined as follows:

The British Thermal Unit is the quantity of heat required to raise the temperature of one pound of pure water one Fahrenheit degree.

When extreme accuracy is desired it is necessary to specify the point on the temperature scale at which the one-degree rise

takes place, as it requires slightly different amounts of heat at different temperatures. This temperature is usually taken either at 39.1° F., at which water has maximum density, or at 62° F.*

For ordinary engineering purposes, however, it is customary and sufficiently accurate to consider the heat corresponding to one-degree rise as constant throughout the scale. Hence the definition given serves for the engineer.

What has been termed a "**mean B.t.u.**" is also used. It is defined as $\frac{1}{180}$th of the heat required to raise the temperature of one pound of pure water from 32° to 212° F. The difference between this mean B.t.u. and the one defined is negligible in most engineering computations.

Sources of Heat.

4. Solar Heat. Heat for human use probably all comes, directly or indirectly, from the sun. This heat is applied directly to produce a sufficiently high temperature on portions of the earth's surface to render plant growth and animal life possible.

Heat engines have been built which convert heat derived directly from the sun into mechanical energy; but, because their bulk is great in proportion to the energy transformed, and because the sun's rays are not always available when needed, they have not as yet been commercially successful.

The energy of the sun's rays is applied indirectly through the agency of plant growth and geologic processes to produce stores of fuel in the earth's crust. Heat energy, indirectly from the sun, may be evolved for human use from this fuel.

Also the sun's rays falling upon water surfaces cause evaporation whereby heat is converted into mechanical energy. This energy lifts the water vapor, which is again condensed and falls upon the earth's surface as rain or snow. The resulting water flowing to its original level prepares the soil for plant growth; it irrigates plants and turns water wheels to supply mechanical energy.

Heat may be derived from mechanical energy, electrical energy, or from the chemical combination of certain elements. In most cases the ultimate source of the energy is probably the sun.

* The value 59° F., corresponding to the scientist's 15° C., is sometimes used.

5. Heat from Mechanical Energy. Primitive man generated heat to kindle fires by rubbing two sticks together. The mechanical energy due to the muscular effort that moves the sticks reappears as heat. This heat is derived indirectly from the sun, since the sun's energy makes possible animal life and therefore muscular effort.

The engineer is familiar with the production of heat by machine friction. This again is a case of conversion of mechanical energy (indirectly from the sun) into heat. This is an undesirable conversion, since mechanical energy, which should be available for useful purposes, becomes useless heat. The same change occurs when a machine is retarded or stopped by a friction brake. This is a useful change, however, since mechanical energy, which cannot be used and which may become dangerous, is dissipated as heat and rendered harmless.

In general, heat for human use is not derived from mechanical energy because it may be obtained in other ways more conveniently and at less cost.

6. Heat from Electrical Energy. The conversion of electrical energy into heat is illustrated by every electric conductor that carries a current; for, though the reason is unknown, heat results whenever an electric current flows. This is known to the electrician as the I^2R-loss. It is always a loss if the delivery of maximum electrical energy is the object of the flow; but it would not be a loss if heat were the object, as in electric furnaces and stoves. Except for special service electrical energy is too expensive a source of heat.

7. Heat from Chemical Combination. There is almost always a liberation of heat when substances combine chemically. In general, the more violent the reaction and the more stable the compound formed, the greater the amount of heat liberated. There are a few combinations which are accompanied by heat absorption; but the compounds formed are generally quite unstable at ordinary pressures and temperatures and the heat absorbed is usually quite small.

The physical chemist briefly explains the phenomenon of heat evolution during chemical combination by saying that chemical energy is converted into heat energy. It is well, however, to

understand his more exact expression, which is at times useful to the engineer.

In every chemical system, or group of systems, there is a certain total amount of "intrinsic" energy. This amount depends upon the kind of system and upon the physical condition. If several such systems react to form new systems, the latter may have a different total quantity of intrinsic energy from the former. If the intrinsic energy of the new system is less than that of the other, energy must have been liberated; if greater, energy must have been absorbed. Energy thus liberated may appear in one or all of its forms; but the largest part of it usually appears as heat.

Thus to supply heat by chemical combination it is necessary to utilize systems that can react to form new systems with less total intrinsic energy.

To illustrate, consider the production of heat by the combination of carbon and oxygen to form carbon dioxide. The total intrinsic energy of the system of carbon molecules and of the system of oxygen molecules is greater than the intrinsic energy of the resulting system of carbon-dioxide molecules. This difference is the source of most of the heat energy used by the engineer.

It is convenient when dealing with these changes to refer to gaseous materials as standards, since the laws of gases are simplest. If a unit weight of gaseous carbon could be combined with gaseous oxygen at some standard temperature and pressure to form gaseous carbon dioxide, a certain amount of energy would be liberated. If some or all of this energy appeared as heat, and if the reacting substances were insulated so that no heat could leave the system, the resulting carbon dioxide would be raised to a high temperature and possibly to a high pressure. Then if heat were withdrawn until the original temperature and pressure were reached, the heat removed might be called the standard heat quantity due to this reaction. Gaseous carbon, however, cannot be used in these operations, only the solid forms being available. Experience and experiment show that to change a solid to a liquid or a gas requires an expenditure of heat. In the chemical combination just referred to, heat is absorbed to change solid carbon to gaseous carbon, so that the heat liberated is less than if gaseous carbon had been used; that is, the heat

liberated is less than that which has been called the standard quantity. Similarly, if the product of the reaction were liquid or solid at ordinary temperatures, instead of a gas, the heat withdrawn to condense to the liquid or solid form would be added to the standard quantity.

There will be further discussion of these phenomena under the head of combustion. They are mentioned here to indicate the nature of the engineer's problem of heat generation.

CHAPTER II.

ELEMENTARY LAWS OF HEAT ENERGY.

8. Conservation of Energy. (a) It seems to be one of nature's great universal laws that energy cannot be created or destroyed. Experience and experiment have tended to establish this law, and now there is no reason to doubt that it holds throughout the universe. This **Law of Conservation of Energy** may be stated as follows: *Energy cannot be created or destroyed; but all forms of energy are mutually interconvertible.*

Unfortunately, the engineer has adopted no unit for the measurement of quantities of energy that is common to all its forms. Each kind of energy is measured in its own unit of quantity, and each unit was originally fixed independently, because of convenience of measurement. The necessity for conversion of units of one form of energy into the units of another form was disregarded; as a result the constant for conversion is sometimes inconvenient for use. Thus, for example, the unit of mechanical energy, the foot-pound, is about $\frac{1}{778}$ of the unit of heat, the British thermal unit; while the unit of electrical energy, the joule, is equal to 0.7373 foot-pounds.

The engineer who deals with heat engines is chiefly concerned with the interconversion of units of heat and units of mechanical energy; he must constantly use the corresponding conversion factor. The determination of this factor requires very accurate experimentation with very delicate apparatus, and the most careful determinations yet made leave some uncertainty as to the exact value. Pending more exact knowledge, engineers commonly use the value 778.

(b) The Law of Conservation of Energy, when limited to heat and mechanical energy, is called the **First Law of Thermodynamics,** and it may be stated thus:

Heat and mechanical energy are indestructible and interconvertible.

The relation of units is

1 B.t.u. = 778 foot-pounds.

In heat engines all of the energy supplied as heat does not appear as mechanical energy. This is not because heat energy is destroyed, but because part of it escapes conversion and leaves the engine still in the form of heat. However, each B.t.u. that is converted is transformed into 778 foot-pounds of work.

(c) In order to do mechanical work there must be motion, and in all real cases the motion meets with resistance of some form. Anything that resists motion takes away energy; thus, friction might take away heat; a belt might take away mechanical energy; a metallic circuit might take away electrical energy; if the motion produces sound, energy is taken away as sound waves in the air. If any energy whatever were taken away, that is, if there were any resistance, and the machine continued in motion without continued energy supply, it would have to give out energy that it did not receive.

It is, of course, impossible to conceive of, or build, a machine which will create energy. Such a machine would give one type of "perpetual motion." To distinguish this type, in which energy is *created*, from the others to be considered later, it will be called **Perpetual Motion of the First Type.**

It follows directly, from the law of conservation of energy, that *Perpetual Motion of the First Type is impossible.* It is also apparent that the First Law of Thermodynamics is a special case falling under this broad general statement.

9. Ideal Mechanisms. In the discussion of some engineering problems it is customary to assume ideal mechanisms for purposes of comparison. There are three types of Perpetual Motion used in discussing these. The first has just been considered; that commonly termed the "Second Type" will be more easily understood later in the discussion. The **Third Type of Perpetual Motion** is that most commonly assumed for purposes of analysis of mechanical problems. It is the ideal perpetual motion of a frictionless machine, which, once started, would continue in motion forever unless stopped by some external resistance or force.

As a matter of fact, no real machine can be frictionless, and therefore no real machine could continue in motion indefinitely; but the friction losses in machines can be reduced to almost negligible values, and for the purpose of analysis this may be

assumed to be carried to the limit, giving perpetual motion of the third type as an ideal possibility.

10. The Second Law of Thermodynamics. (a) It is a matter of common observation that in a steam engine, for instance, the steam exhausted still contains a considerable quantity of heat, and that its temperature is lower than that of the steam supplied to the engine.

This phenomenon of receiving heat at a high temperature and rejecting some of it at a lower temperature is characteristic of every real engine, and will be shown later to be characteristic of every ideal engine, no matter how perfect. The operation of all such engines is pictured graphically in Fig. 1. Heat energy at the high temperature T_1 flows from reservoir *I* into the engine. There part of it is converted into the stream of mechanical energy (shown flowing out to the right), while the rest passes completely through the engine and emerges, still in the form of heat, but at the lower temperature T_2 of heat receiver *II*, which absorbs it.

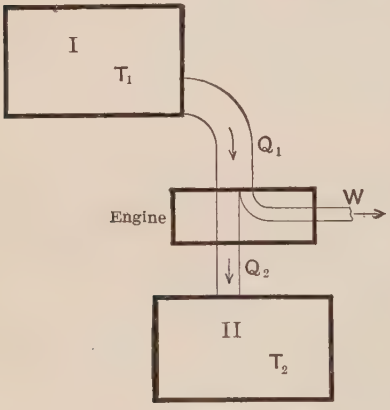

Fig. 1 — Diagrammatic Representation of a Heat Engine.

Calling the heat supplied in a given time Q_1, the mechanical energy leaving W, and the heat leaving Q_2, it follows from the conservation of energy that

$$W + Q_2 = Q_1.$$

This rearranged gives

$$W = Q_1 - Q_2,$$

from which it immediately appears that the smaller Q_2 is the greater will be the work resulting from the use of a given quantity of heat Q_1.

(b) Experience has shown that no device can even be imagined which, under existing circumstances, could continuously convert into mechanical form all of the heat energy supplied it. All machines so far devised, actual or ideal, can continuously convert

only part of the heat supplied them and must *reject the remainder at a lower temperature* than that at which it was received. This is summed up in the so-called **Second Law of Thermodynamics** as follows:

*No machine, actual or ideal, can both completely and continuously transform heat into mechanical energy.**

(c) If such *complete* transformation could be effected, it would give what is called **Perpetual Motion of the Second Type.**

So long as heat must be exhausted at a lower temperature the possibility of obtaining mechanical energy from heat ceases as soon as the temperature of all the heat in the universe has been dropped to the lowest attainable value.

If this necessity of exhausting heat at a lower temperature were removed, it would be possible to continue the conversion of heat into mechanical energy after all means of obtaining a temperature difference had been used up, that is, after all heat had been reduced to the lowest attainable temperature.

As all mechanical energy eventually passes back into heat energy (generally at low temperature) through friction and allied phenomena, there would be no danger of the supply of heat giving out. The cycle would then be an endless one, consisting of the transformation of heat into mechanical energy, the retrogression from this form of energy to heat, the conversion to mechanical form again, and so on *ad infinitum.*

This would then be equivalent to a sort of perpetual motion which is distinguished from the other two types by calling it, as above, Perpetual Motion of the Second Type. Hence the Second Law of Thermodynamics may also be stated thus:

Perpetual Motion of the Second Type is impossible.

This Second Type of Perpetual Motion, like the First Type, is impossible even in imagination, whereas the Third Type, though impossible of realization, is an ideal limit of possibilities.

11. Distribution of Associated Heat Energy. (a) Common experience shows that the quantity of heat associated with any portion of matter may be changed and that the transformation is accompanied by other definite phenomena such as change of

* There are almost as many statements of the Second Law as there are authors of books on thermodynamics. It is believed that the statement as here given is the most satisfactory for the purposes of this book.

pressure, or of volume, or of temperature, or of physical state; chemical change may also occur and other forms of energy may appear or disappear.

Despite these varied possibilities a very simple and definite generalization may be made. This at least serves the purpose of establishing a viewpoint and aids in analysis.

Consider, for example, a single chemical substance, which may be an element or a compound, and which is assumed *not to be set in motion as a whole, nor to be altered chemically, nor to lose energy by any form of radiation.* In such a substance there can be only three results from adding heat, and there are only three sources from which heat can be abstracted.

1. Heat addition may be accompanied by **rise in temperature.** In this case that part of the heat which is used in causing the temperature change may be conceived as effecting an increase in the motion of the constituent particles. Heat thus used is known as **Sensible Heat** and its addition increases the substance's store of sensible heat. Conversely, the abstraction of heat may be accompanied by a fall in temperature (probably decrease of internal motion), and the source of part of the abstracted heat is the store of sensible heat of the substance.

2. Heat addition may be accompanied by a **variation of the internal structure** of the substance, and this may be imagined as a molecular rearrangement. The part of the heat which is used in causing this change is called **Internal Latent Heat.** Conversely, the abstraction of heat may cause the reverse change.

3. In 1 and 2 when heat is added the size of the substance may change; there would then be a positive or negative **displacement of surrounding media, against resistance,** and part of the heat added supplies the necessary mechanical energy for this displacement. The heat thus transformed is sometimes called **External Latent Heat.** Conversely, when heat is abstracted in 1 and 2 the surrounding media may return and the equivalent of the external latent heat be abstracted as heat.

(b) Heat energy added may then be imagined to produce results as follows:

In 1, part of the added heat may increase the kinetic energy of the molecules.

In 2, part of the added heat may overcome the resistance to rearrangement of the molecules.

In 3, part of the added heat may overcome the resistance of surrounding media to displacement.

In all cases the added heat becomes stored energy; for if the phenomena are reversed (conduction and radiation loss being prevented) the substance will return to its original dimensions, state, and temperature, and the energy previously given to the substance to accomplish these results will be returned as heat. Since the sensible heat and the internal latent heat are stored *within* the substance itself, and since the external latent heat is stored in *external* media, it is common to call the sum of the first two the **Change of Intrinsic Heat Energy** and the third the **Change of External Heat Energy.**

(c) The following symbols will be used to designate, **in thermal units**, the various quantities concerned in changes of associated heat energy in substances:

ΔQ = the total quantity of heat added to or taken from the substance.

ΔS = the part of ΔQ associated with temperature change; this equals the change of sensible heat.

ΔI = the part of ΔQ associated with internal rearrangement; this equals the change of internal latent heat.

ΔE = the part of ΔQ associated with the displacement of external media; this equals the change of external latent heat.

From the foregoing discussion it follows that:

$$\Delta Q = \Delta S + \Delta I + \Delta E, \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

for the three symbols on the right of the equation represent the only destinations possible for added heat, and the only possible sources of abstracted heat.

Thus the change of intrinsic heat energy $= \Delta S + \Delta I$, the change of external heat energy $= \Delta E$ and the change of total associated heat energy $= \Delta Q = \Delta S + \Delta I + \Delta E$.

It is in general possible for any or all of the three terms on the right of the last equation to be either positive or negative or equal to zero. Hence it is necessary, within the conditions set at the beginning, to consider the equation as perfectly general, and to interpret it for the conditions of each case.

(d) As an illustration of the foregoing statements, consider the transformations that occur and the heat that is utilized

in generating steam from cold water. In the cylinder, above burner *a* in Fig. 2, let there be cold water, say at room temperature, below the piston. Let the temperature be raised by the external application of heat.

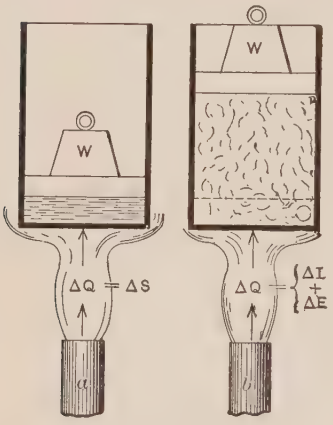

Fig. 2.

Common experience shows that this rise of temperature will be accompanied by a slight increase of volume, and refined experiment leads to the belief that it will also be accompanied by certain intramolecular changes. Because of the volume increase some of the heat supplied in raising the temperature must be used in doing the *external work* of lifting the weight W and the piston against the action of gravity, and of moving the piston against the atmospheric pressure on its upper side. This part of the total heat supply may be called ΔE. Because of the intramolecular work and because the molecules must be separated against any interattractions, as the volume increases, some of the heat applied during the temperature increase must be used for doing *internal work* and may be designated by ΔI. The part of the heat supplied and not accounted for by the sum of ΔE and ΔI must be that used in what is recognized as a rise of temperature and may therefore be designated by ΔS, the *sensible heat.*

Then the total heat (ΔQ) supplied during the temperature rise is $\Delta Q = \Delta S + \Delta I + \Delta E$.

It so happens, however, that in this case, in which water is heated, the numerical values of ΔI and ΔE are so very small, as compared with that of ΔS, that for engineering purposes they may be neglected without serious error and ΔQ may be taken as equal to ΔS, as illustrated at *a* in Fig. 2.

(e) Suppose, now, that the temperature of the water in the cylinder has been raised to that at which steam is formed. Then continued addition of heat will not further raise the temperature, but it will cause the formation of steam (at constant temperature and constant pressure), with a very great increase of volume,

and its accompanying separation of molecules. There will also probably be certain intramolecular changes.

As there is no temperature change during the process of vaporization, no part of the heat (ΔQ) supplied can be used to change the sensible heat, that is, as ΔS.

The enormous increase of volume during vaporization, with the consequent raising of the piston against the resistance offered by the weight of the piston, the superincumbent atmosphere, and the weight W, involves the doing of *external work*, and some of the heat supplied during the process must be used for that purpose. This heat, which may be designated by ΔE, is known as the *external latent heat* of vaporization and is stored as potential energy in the mechanical parts of the system, *not* in the steam itself.

The intramolecular and the intermolecular work consume the rest of the heat supplied, and the part used for such purposes is called the *internal latent heat* of vaporization. According to the symbols adopted it would be designated by ΔI.

Thus the heat supplied during vaporization is

$$\Delta Q = \Delta I + \Delta E,$$

and this is shown at b in Fig. 2.

Considering the whole process of heating the water and vaporizing

$$\Delta Q = \Delta S_l + \Delta I_l + \Delta E_l + \Delta I_v + \Delta E_v$$

in which the subscript l indicates heat added to the liquid while raising the temperature and subscript v refers to the heat added during vaporization. On the assumption that ΔI_l and ΔE_l are negligible,

$$\Delta Q = \Delta S_l + \Delta I_v + \Delta E_v.$$

While water has been used as an example, all liquids present similar phenomena during heating and vaporization. Liquid ammonia, liquid sulphur dioxide, liquid carbon dioxide, or any one of a number of other materials, might have been used as an illustration.

Other examples of processes showing the different utilizations of heat might be cited, but it is believed that, for present purposes, the one given above sufficiently illustrates the ideas and the meanings of the symbols used.

12. **Specific Heat.** (a) As just indicated, the change of temperature, with corresponding change of sensible heat, may be accompanied by two other changes, and it is clear that the change in associated heat energy is dependent upon all three factors. This must be taken into account in considering **specific heat,** which may be defined thus:

The specific heat of a substance is the heat added to, or abstracted from, a unit weight of that substance when its temperature is changed one degree.

The quantity of heat thus defined may be used in any one or all of three ways: (1) to raise temperature, (2) to do internal work, (3) to do external work. The quantity of heat required simply to raise the temperature would obviously be less than the quantity required to raise the temperature and also to do work, external or internal. Hence for every substance there must be several specific heats, the values of which depend upon the use made of the heat.

But by whatever method the heat is applied and whatever the use made of it during its addition to a substance, if the method is the same throughout, the specific heat, C, by definition must be

$$C = \frac{\Delta Q}{W(T_2 - T_1)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

in which

ΔQ = heat added.
W = weight of substance receiving heat.
T_1 = temperature before ΔQ is added.
T_2 = temperature after ΔQ is added.

(b) If the specific heat is *not constant*, with any method of heat application, the value of C from Eq. (2) is an *average* value for the temperature range and is called a **Mean Specific Heat.***

Hereafter mean specific heats will be denoted by putting a vinculum over the symbol, and, where essential, the temperature range will be indicated by subscripts; thus, $\overline{C}_{75-180}$ should be read as the mean specific heat between 75° and 180°.

If the specific heat is *constant* the heat added during the temperature change from T_1 to T_2 is

$$\Delta Q = CW(T_2 - T_1) \quad . \quad . \quad . \quad . \quad . \quad (3)$$

* The mean specific heat is thus useless for purposes of exact calculation unless the temperature range over which it is the average is known and unless it is used in calculations involving that same temperature range.

If the specific heat is *variable*

$$\Delta Q = \bar{C}W(T_2 - T_1). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4a)$$

or

$$= W\int_{T_1}^{T_2} C\,dT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4b)$$

in which C represents the successive, or instantaneous, values of the variable specific heat as the temperature changes from T_1 to T_2.

(c) It is conceivable that the temperature of a substance may be raised in such manner that no internal or external work is done, and the heat would then be applied only to raising temperature, and would be a **true specific heat.** Such true specific heats, it will be found later, are sometimes closely approximated in the case of gases.

13. Total Associated Heat. It is impossible at present to determine the *total* quantity of heat energy associated with a substance under given conditions, because no means are available for complete heat removal.

To compare associated heats of substances at different temperatures, a convenient value for T_1 is assumed as a datum and calculations are confined to the region above it. This method gives relative and not absolute results, but serves for engineering purposes. The value of T_1 is usually 32° F., when such a choice is possible.

CHAPTER III.

THE HEAT-POWER PLANT.

14. General. It has already been stated that the heat-power engineer has to do largely with the conversion of the latent heat in fuels into available mechanical energy. This transformation is effected by means of various pieces of apparatus, used singly in some cases and in series in others. All of the apparatus necessary in any individual case may be called a "Power Plant."

That part of the power plant which receives heat energy and delivers mechanical energy, i.e. the "engine," is often called a **"Prime Mover."**

Examples of heat-power plants are familiar to all. They may contain Steam Engines and Boilers, with certain auxiliary apparatus necessary for the satisfactory operation of these two principal pieces; they may contain Gas Producers and Gas Engines, with suitable "auxiliaries;" or they may contain an engine only, as in the case of a gasoline-engine or an oil-engine power plant.

No matter what the type of plant, a certain general method of operation is common to all. This is illustrated in Fig. 1. Heat, from some kind of fuel, is continually flowing in, forming a "stream of energy." This energy leaves the system in a number of different ways, divisible broadly into waste (or lost) energy and useful energy. In all plants, including those theoretically perfect, there must always be a loss; thus the energy flowing out in useful form must always be only a fraction of that flowing in. This will be illustrated by the description of the operation of the Steam-Power Plant, in Section 15, and of the Producer Gas-Power Plant, in Section 16.

15. The Steam-Power Plant. (a) This type, which is the oldest and is the most used of all forms of heat-power plant, may be said to consist of four essential parts, — the "Steam

Boiler," including the "furnace," the "Steam Engine," the "Condenser," and the "Boiler Feed-Water Pump." It is shown in one of its many forms in Fig. 3, with these four parts named. The method of operation is as follows.

(b) Fuel is burned on the "grate" in the furnace under the boiler. The combustion of this fuel liberates a large amount of heat energy, which is partly absorbed by the products of combustion, partly radiated to the water through the "heating surfaces" of the boiler tubes and shell, and partly radiated through the furnace walls to the surrounding air. This latter type of radiation represents a loss, which can never be prevented, as the furnace walls cannot be made nonconducting.

The furnace may be regarded as the part of the boiler apparatus which converts the heat energy, latent in the fuel, into available heat energy. It will be found that there are certain losses in this conversion which can never be entirely prevented in any real case. They may be summarized as follows:

(1) Some of the fuel falls through the grate and is not burned.

(2) The ashes and refuse drop through the grate with a higher temperature than that at which they were put into the furnace.

(3) Some of the more volatile parts of the fuel pass off with the products of combustion and are not burned.

(4) In order to insure the complete combustion of the fuel, a larger amount of air must be supplied the furnace than is theoretically necessary. This mixes with the products of combustion proper and represents just so much more gas to be heated by the energy liberated by combustion. As a result the temperature attained by these gases is proportionately lower and, as will be discovered later, the subsequent utilization of the heat is made more difficult.

(c) During the operation of the furnace a stream of radiant heat energy from the incandescent fire passes through the heating surfaces to the water and steam, and there is also a stream of hot gas which, as it passes over these surfaces, gives up heat to the fluids within. In this part of the process there will always be three losses:

(1) Part of the heat carried by the products of combustion will pass out through the external walls of the "boiler setting," instead of into the heating surfaces.

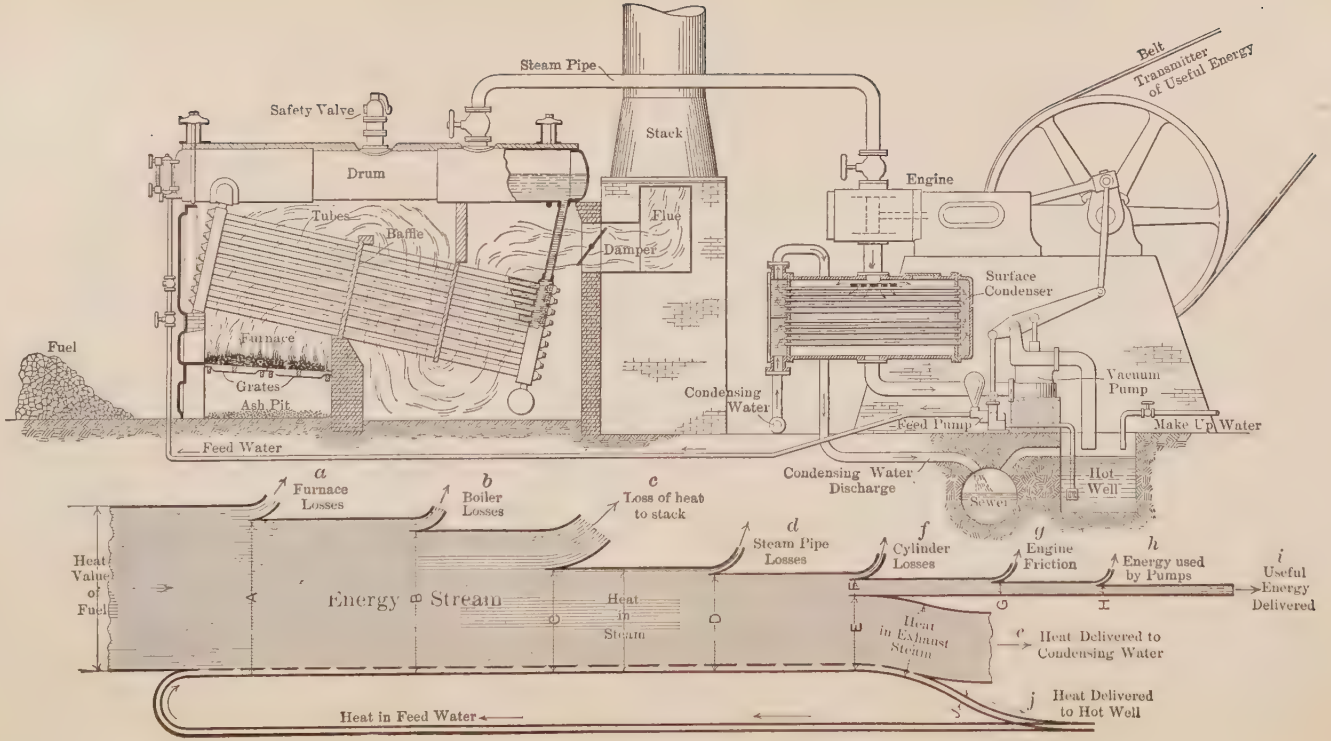

Fig. 3. — Elements of a Steam-Power Plant.

(2) The gases can in theory pass heat into the heating surfaces so long as their temperature is higher than that of the water and steam. In the ideal boiler these gases would be cooled to the temperature of the water and steam, but in practice, for various reasons, they leave the apparatus when there is still a difference of temperature of from 200° F. to 500° F. or even more.

(3) The temperatures of the fuel and air entering the furnace of course approximate that of the room, and the products of combustion are heated from that value (about 60° F.) to the high temperature with which they leave the furnace. The temperature of the water and steam within the boiler is always from about 200° to 400° F., or more, higher than room temperature; thus, even if the gases were cooled the theoretically maximum amount, they would still carry off considerably more heat than they would if cooled to atmospheric temperature.

Despite all the losses so far enumerated, a considerable proportion (from 50 to 80 per cent) of the original heat energy of the fuel is passed through the heating surface and is used in raising the temperature of the water and in generating steam. This heat is stored in the steam.

(d) As was explained in Section 11 (d), under the conditions governing the generation of steam in a steam boiler, the liquid must first be raised to a definite temperature, dependent on the pressure, before it can be vaporized. To raise the water to this temperature a certain amount of heat must be added to it; and, the lower the temperature at which the water enters the boiler and the higher the temperature of the steam, the greater will be the quantity of heat needed.

After the absorption of this amount of heat a still larger quantity, known as the "latent heat of vaporization," must be supplied to convert the hot water into steam at the same temperature.

The total heat supplied can be subsequently abstracted, as heat, by condensing and cooling and thus obtaining the same water at the original temperature, or part of it can be obtained in the form of useful mechanical energy by certain transformations which may be made to take place in the steam-engine cylinder. For our present purposes this latter is the more important of the two possibilities.

(e) The steam is led to the cylinder of the steam engine by the "steam pipe" shown in Fig. 3. In the cylinder a part of the heat in the steam is converted into mechanical energy by the action of the steam on the piston, part is wasted by "cylinder losses," which will be considered in a later chapter, and the remainder is still in the steam when this is discharged from the cylinder. Only from 5 to 22 per cent of the heat available in the steam is converted into useful energy in the cylinder, and, because of friction of engine parts and work done in driving the pumps, not all of this is delivered by the engine to the belt or other power consumer.

The exhaust steam, still retaining the greater part of the heat that was furnished by the fuel, is conducted to the "surface condenser" (see Fig. 3), where it is condensed on the outer surfaces of the condenser tubes, through which the "condensing water" is circulated. The heat which the condensing water absorbs in liquefying the steam is the "latent heat of vaporization," and this is large in amount. In the plant shown this heat is not further utilized and hence represents a considerable loss.

The water resulting from the condensation of the steam, known as the "condensed steam" or "condensate," is transferred from the condenser, in which the pressure is below that of the atmosphere, to the "hot well" by the "vacuum pump," which also removes any air which may accumulate in the condenser. The "feed-water pump" takes this water from the hot well, with whatever heat it contains, raises its pressure to that of the steam, and returns it to the boiler, where it is reconverted into steam and started again on the round just described.

(f) Such a combination of processes, which periodically brings the material back to starting conditions, is known as a **"cycle,"** or, more properly, as a "closed cycle."

The characteristic of the cycle above outlined is the fact that the water (or, speaking more generally, the **"working substance"**) is not lost or used up or permanently altered in any way. It serves simply as a carrier and transformer of heat energy, receiving it from the boiler furnace, giving up some of it as mechanical energy to the piston, rejecting the remainder to the condenser, and then returning to the boiler to start the cycle once more.

This is, in theory at least, characteristic of all processes by means of which heat is converted into work. In practice it is sometimes found to be simpler or more desirable to throw away the working substance after it has been used in the engine and to continue to supply new quantities for the reception of heat at high temperature. In steam-power plants, for instance, the condensate is often abandoned, and the boilers are then supplied with corresponding quantities of water from some other sources, such as wells or streams.

Theoretically, however, it is immaterial whether one pound of water is used time after time, or whether new is substituted for old, pound for pound, at some point in the process, provided only that the substitute have the same volume, pressure, temperature, and heat conditions as that which it replaces.

In the lower part of Fig. 3 is a "heat-flow diagram." This shows the stream of heat energy flowing from the boiler to the engine. Its width shows the relative amount of heat remaining available for doing external work, and the offshoots show the losses that occur at different stages of the process.

(g) Returning to the engine, the action of the steam within the cylinder will now be considered in a very elementary manner, in order to bring out certain conceptions which will be useful in the discussions of the following chapters. This action of the steam will be considered in detail later.

The steam, upon its arrival at the engine (which, for simplicity, will be considered "single-acting"), is admitted by the "admission valve" to one end of the cylinder, where it acts on the piston, causing it to move and deliver mechanical energy. The valve may remain open during the entire stroke of the piston, or it may close before the stroke ends, which is the usual practice.

If the different positions of the piston in its stroke are plotted as abscissas and the corresponding pressures acting on the piston face are erected as ordinates, there will be obtained a line like *ab*, in Fig. 4, which line is a graphical representation of pressures occurring within the cylinder during the admission of the steam. The work done on the piston in moving it from position 1 to 2 can be computed if the constant pressure acting on the piston and the distance traversed are known.

If the admission valve be assumed to close (at 2) when the distance moved is less than the stroke, the expansion of the steam

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thus entrapped will continue to drive the piston until the end of the stroke is reached. It will be discovered in later chapters that this expansion is accompanied by a drop in the pressure and in the temperature of the steam. The way the pressure drops during the expansion is shown by the curve *bc* in Fig. 4. The work done while the piston is moving from position 2 to 3 can be computed when the average pressure acting on the piston and the distance traveled are known. The average pressure is proportional to the mean ordinate of the curve *bc*. The method of determining it will be given later. At present it is sufficient to know that work is done during this expansion.

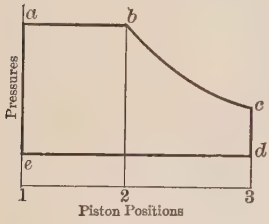


Fig. 4.

When the piston arrives at the end of its stroke the " exhaust valve " opens communication between the interior of the cylinder and the condenser, in which a comparatively low temperature and pressure are maintained, and some of the steam at once rushes from the cylinder to the condenser, where it is liquefied and discharged to the hot well in the manner already discussed. This process is shown by the line *cd* in Fig. 4, the pressure within the cylinder decreasing to the value prevailing within the condenser.

If, now, the piston is driven back to the beginning of the stroke, it will force all the steam from the cylinder into the condenser, where it will be liquefied as fast as it enters. In moving from position 3 to 1, in Fig. 4, the piston will have swept through the entire stroke against a constant resisting pressure equal to the " back pressure " or " condenser pressure." This operation is represented by the line *de* in Fig. 4. In forcing the steam from the cylinder, the piston does work which can be computed if the mean resisting pressure, as shown by the ordinate of *de*, and the stroke are known.

Obviously, no work is done on or by the piston during the process represented by *cd*, as it is not accompanied by motion of the piston. Similarly, if the piston is stationary at the beginning of its stroke while the pressure is raised by the entering steam from the value shown at *e* to that at *a*, no work is done during that process.

The *total* work done during the two strokes is the difference between the work done *on* the piston during processes represented by lines *ab* and *bc* and that done on the steam *by* the piston during process *de*.

The processes through which the steam has been carried in the cylinder (as shown in diagram, Fig. 4) are idealized versions of what occurs in actual engines, and it is seen that even in this ideal case only a small part (theoretically, from 10 to 30 per cent) of the heat in the steam could be actually converted into work, the rest remaining in the steam exhausted. In the actual case a still larger amount of the heat is wasted in the cylinder and a proportionately less amount of heat is delivered as mechanical energy to the piston. These losses occurring within the cylinder are quite large and may be called " cylinder losses."

Not all the mechanical energy that is available at the piston is delivered by the engine (by belt or other means) for doing useful work, for some of this energy is used in overcoming the friction of the engine itself.

The relative amounts of energy available for doing work and the losses occurring at the different stages are shown in amount by the width of the energy stream in the lower part of Fig. 3.

(h) In imagination at least, it may be considered that the same operations that have been described for the power plant as a whole can be performed entirely within the engine cylinder alone. Thus the water (or working substance), constant in amount, can be considered as always remaining within the cylinder, and can be imagined first as being there heated and vaporized (corresponding to lines *ea* and *ab* in Fig. 4), then as steam acting on the piston during the expansion (corresponding to line *bc* in Fig. 4), and finally as being condensed and returned to its original condition (according to lines *cd* and *de*) by the abstraction of heat by some process equivalent to that performed by the condenser.

As all these processes are imagined to be performed with the same working substance, and as this is always returned to its original condition, the operations within the cylinder may be said to constitute a cycle, which may be called " the engine cycle " to distinguish it from the cycle of the power plant as a whole.

(i) It should be noted that, in obtaining mechanical energy from heat by means of the engine, the working substance supplies

heat energy to the engine at a high temperature (that is, it furnishes what may be called "high-temperature heat"), and that upon leaving the cylinder the working substance still retains some of the heat but at a lower temperature (that is, it retains what may be called "low-temperature heat"). This will be found to be characteristic of every process by which heat is converted into mechanical energy. Evidently, the more heat converted into mechanical energy and the less rejected at low temperature, the more efficient is the engine. But even in the ideal case it will be found that some heat must be rejected, which is in accordance with the statement of the Second Law of Thermodynamics and is shown diagrammatically in Fig. 1.

16. The Producer Gas-Power Plant. The principal parts of this plant are represented in Fig. 5. The fuel enters the gas producer, carrying with it its store of heat. In the producer the combustible part of this fuel is gasified at the expense of some of its heat, while in theory the rest of its heat is stored in the combustible gas formed. The gas, carrying this part of the heat with it, enters the engine cylinder, mixes with air, and is ignited. The resulting inflammation raises the temperature and pressure of the products of combustion to high values. These gases then do work on the piston at the expense of this high-temperature heat and sustain a corresponding drop in temperature. They are finally rejected, carrying with them a certain part of the original heat content, now existing at a lower temperature.

Theoretically, it would be possible to remove this low-temperature heat in an apparatus corresponding to a condenser, return the same mass of working substance to its original chemical composition, and start the cycle over again. Practically, however, it is found much simpler to throw the burned gases away each time and to start again with fresh working substance.

For this reason the atmosphere is commonly used in place of a condenser. It possesses the necessary characteristic of low temperature, as compared with the highest attained in the operation of the engine, and has ability to absorb all the heat rejected by the engine. It possesses the further convenient characteristic of being able to absorb the working substance as fast as it is rejected by the engine.

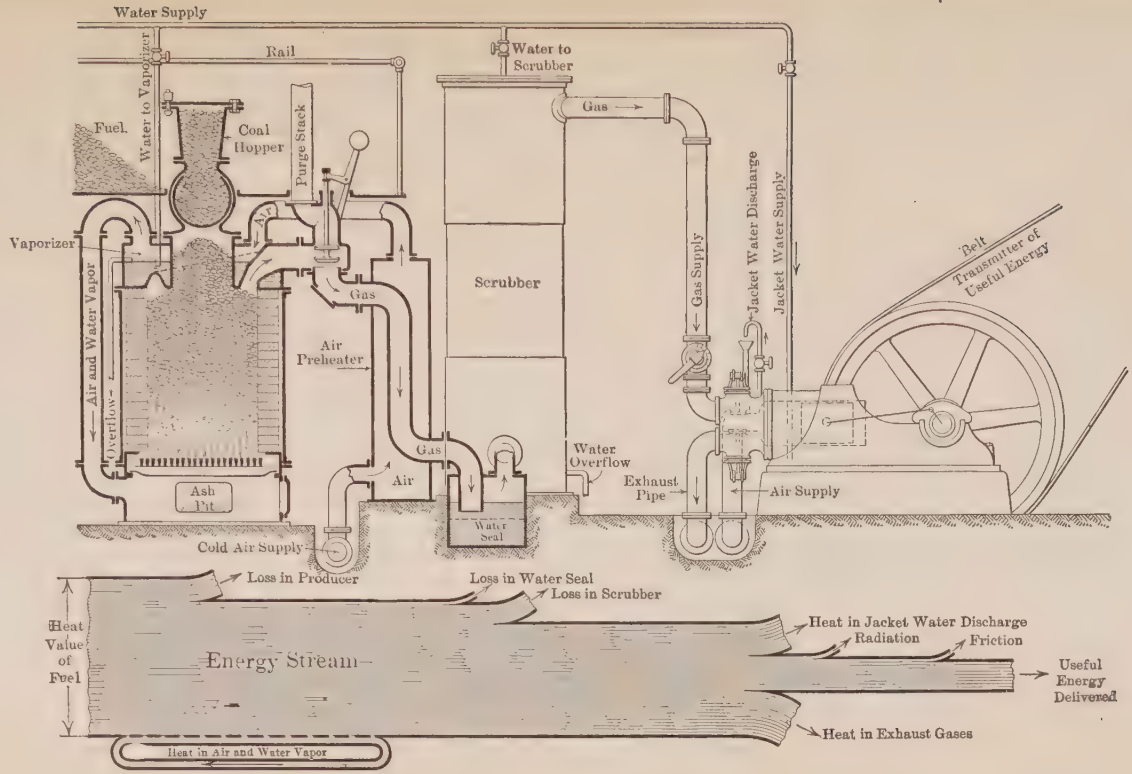


Fig. 5.—Elements of a Producer-Gas Power Plant.

Although the series of operations that was outlined in connection with the steam-power plant is not quite so evident in this case, analysis will show that, in theory at least, the working substance could be used over and over again, serving only to receive high-temperature heat, to transform some of it into mechanical energy, and to reject the rest at a low temperature.

17. Analogy. The operation of heat engines has often been compared to the operation of water wheels, and there is much that is similar.

A water wheel develops mechanical energy by receiving water under a high head, absorbing some of its energy, and then rejecting the fluid under a low head.

A heat engine, in developing mechanical energy, receives heat energy at a high temperature (head), absorbs some of it, which is converted into mechanical energy, and then rejects the rest at a low temperature (head).

This analogy between "heat sliding down a temperature hill," as it is sometimes stated, and "water sliding down a grade," is very useful, but should not be carried too far.

One point of resemblance is, however, worthy of special note: The water wheel never removes all of the energy of the water; there is always a certain discharge loss, or a certain amount of energy rejected. In the same way the heat engine never removes all of the heat energy from the working substance; there is always a certain discharge loss, or a certain amount of energy rejected (Second Law of Thermodynamics).

18. Further Study. A number of theoretical considerations must be studied in detail before this subject of conversion of heat energy into mechanical form can be discussed more thoroughly. It is necessary to learn some of the physical and chemical properties of the common working substances, some of the different kinds of changes they can be made to undergo for the doing of work, and to develop certain theoretical cycles of operation upon which the real cycles are based.

This is done in the immediately succeeding chapters. Gases are considered first because their laws permit of simpler forms of expression and are more easily understood than are those of

vapors, which are the only other working substances commonly used.

In later chapters the real cycles, the engines, and their auxiliaries, the power plants, and the commercial and operating considerations connected therewith, will be taken up again.

CHAPTER IV.

THE LAWS OF GASES.

19. States of Aggregation of Substances. (a) Almost every substance known has, under proper temperature and pressure conditions, been made to exist in three physical states, or conditions of aggregation, — namely, as a solid, as a liquid, and as a gas; and it is probable that this can be done for all matter with proper regulation of temperature and pressure.

The higher the temperature and the lower the pressure, the greater the tendency to exist in the more rarefied condition of aggregation — that is, as a gas; while the lower the temperature and the higher the pressure, the greater the tendency toward the solid form. The values of the limiting conditions, — namely, temperature and pressure, — which will determine any of the three states, vary widely with the different substances, and under ordinary atmospheric conditions some of the materials in the universe are known as solids, others as liquids, and still others as gases.

The various substances obey certain laws, differing for different states, and with constants that vary with the substance. The laws that govern matter in the gaseous state are the simplest and at present are best known. These laws will now be developed and will be more fully discussed in Chapter IX.

(b) The laws of gases may be divided into two groups, — Ideal Laws and Actual Laws.

The **ideal laws** or **laws of ideal gases** are not absolutely true for any real gases, but hold, with sufficiently close approximation for engineering purposes, for all gases which are far removed from liquefaction, like hydrogen, nitrogen, oxygen, and, to a certain extent, carbon dioxide.

The **actual laws** of gases are the ideal laws modified so as to conform as accurately as possible to the behavior of **real gases**, and they generally take account of different theories of the

actual composition of gases. They are seldom used by engineers and their consideration is left for another chapter.

In general, the variation from the ideal laws becomes less as the real gases are further removed from the conditions of liquefaction, or as the molecules become more widely separated and the effect of intermolecular forces becomes less. From this it is concluded that the **hypothetical ideal gas** must be imagined devoid of such intermolecular forces.

20. The Ideal Laws of Condition of Gases. These laws are,

1. The Law of Boyle or of Marriotte, and
2. The Law of Charles or of Gay Lussac.

1. Boyle's Law.

This law, which deals with variations of pressure and volume at **constant temperature**, is:

When the temperature of a given weight of gas is maintained constant the volume and pressure vary inversely. Mathematically expressed, it becomes

$$\frac{V_1}{V_2} = \frac{P_2}{P_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

or

$$V_1P_1 = V_2P_2 = V_3P_3 \quad . \quad . \quad = V_nP_n = \text{Constant}, \quad . \quad (6)$$

in which

V_1, V_2, etc. = the volumes occupied by a given weight of a particular gas at constant temperature but different pressures, and

P_1, P_2, etc. = the corresponding pressures exerted by the gas, or to which the gas is subjected.

2. Charles' Law.

(a) This law, which deals with volume or with pressure changes accompanying **temperature variations** may be conveniently divided into two statements:

(1) *When the pressure of a given weight of gas is maintained constant the volume increases* $\frac{1}{491}$* *of its value at* 32° *F. for every Fahrenheit degree rise of temperature and decreases the same amount for every degree decrease of temperature.*

* The exact value is not $\frac{1}{491}$, but this is probably the nearest simple fraction and is close enough for engineering purposes.

(2) *When the volume of a given weight of gas is maintained constant the pressure increases* $\frac{1}{492}$ *of its value at 32° F. for every Fahrenheit degree increase in temperature and decreases the same amount for every degree decrease in temperature.*

(b) Given a unit volume of gas at 32° F. with pressure maintained constant, then increasing the temperature 1° F. would cause the volume to become $\frac{1}{492}$ larger, while a decrease of 1° F. would result in a volume $\frac{1}{492}$ smaller; a 2° change in temperature would cause the volume to alter $\frac{2}{492}$, and so on. Writing temperatures and corresponding volumes for this case side by side, and beginning with a temperature of (492 + 32) degrees, gives:

Temperatures, Fahr.	Volumes
524° (= 32 + 492) . . .	$1 + (492 \times \frac{1}{492}) = \frac{984}{492} = 2$
.	
.	
33°	$1 + (1 \times \frac{1}{492}) = \frac{493}{492}$
32°	$1 + (0 \times \frac{1}{492}) = \frac{492}{492} = 1$
31°	$1 - (1 \times \frac{1}{492}) = \frac{491}{492}$
.	
.	
0°	$1 - (32 \times \frac{1}{492}) = \frac{460}{492}$
.	
.	
− 460° (= 32 − 492) . .	$1 - (492 \times \frac{1}{492}) = \frac{0}{492} = 0.$

If these volumes are plotted as abscissas with temperatures in degrees Fahr. as ordinates, the points will be found to lie on a straight line which intersects the temperature axis at − 460°.

If the law holds consistently the volume will be reduced to zero at − 460° F. Similarly with constant volume the pressure must become zero at − 460° F. This point of the temperature scale is called the **Absolute Zero** of temperature, and temperatures measured from it are known as **Absolute Temperatures.** Since this point is 460 Fahrenheit degrees below Fahrenheit zero, the absolute temperature, T, corresponding to any Fahrenheit temperature, t, can be found by adding 460 to the latter; that is,

$$T = 460 + t. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

(c) The conception of absolute temperature makes possible a very simple **mathematical statement of Charles' law.** This should be evident from the foregoing table.

Thus, the two parts of the law are:

(1) With **pressure constant** $\frac{V_1}{V_2} = \frac{T_1}{T_2}$, ... (8)

and

(2) With **volume constant** $\frac{P_1}{P_2} = \frac{T_1}{T_2}$. . . . . (9)

(d) The apparent anomaly of zero volume at absolute zero temperature results from assuming the law to hold continuously to the lowest temperatures. It should be remembered that this is a law for an *ideal* substance only, and does not represent the behavior of any material actually existing. Therefore there is no *a priori* reason for doubting the result. A possible explanation of this matter will be given in Section 76 (e). Despite its apparently ridiculous meaning at low temperatures, the law holds with sufficient accuracy for most gases at the temperatures used in ordinary engineering.

3. Combination of the Laws of Boyle and Charles.

(a) Since it is seldom true in actual practice that one of the three possible variables, P, V, and T, remains constant while the other two change, it is convenient to combine Boyle's law with that of Charles so as to obtain an expression giving the relation among all three variables. The resulting expression is known as the Law of Condition of Ideal Gases, or, more simply, as the **Law of Ideal Gases.**

(b) To obtain the mathematical expression of this law, it is only necessary to imagine a given weight of gas with initial conditions P_1, V_1, T_1, changing to final conditions P_2, V_2, T_2, in two steps; first, at constant temperature T_1, to V_2 and some intermediate pressure P_1', and second, at constant volume V_2, to P_2 and T_2.

The result of the first change is given by Boyle's law as follows:

With temp. constant at T_1, $\frac{V_1}{V_2} = \frac{P_1'}{P_1}$,

from which $P_1' = \frac{P_1 V_1}{V_2}$. . . . . (10)

Here P_1' is the resulting pressure of the gas when its volume is changed to V_2 and its temperature remains T_1. Then using Charles' law for the second change,

with volume constant at V_2, $\frac{P_1'}{P_2} = \frac{T_1}{T_2}$,

from which $$P_2 = \frac{P_1' T_2}{T_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Here P_2 is the resulting pressure of the gas when its temperature changes to T_2 and its volume remains V_2.

If now the value of P_1' from (10) be substituted in (11), the resultant expression is

$$P_2 = \frac{P_1 V_1}{V_2} \cdot \frac{T_2}{T_1}$$

giving, on rearrangement,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

which is the expression sought. In general this becomes

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3} = \quad . \quad . \quad \frac{P_n V_n}{T_n} = \text{Constant}. \quad . \quad . \quad (13)$$

(c) The value of this **constant** for any given gas will vary directly with the weight of gas dealt with; for, at any given temperature and pressure, two pounds of gas must occupy twice the volume occupied by one pound, three pounds three times the volume, etc. For convenience it is customary to tabulate, for all the commercial gases, the value of this constant obtained by substituting in (13) the volume of *one pound* of gas and the temperature and pressure at which the volume was experimentally determined. This constant, commonly represented by R, will be found to be of great importance.

To distinguish the volume of any given weight from the volume of a *unit* weight of gas, the former will hereafter be designated by V and the latter by **v**. The expression of the **law for one pound** of ideal gas then is

$$\frac{P\mathbf{v}}{T} = R. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

(d) It makes no difference what units are adopted in such an expression as (13), provided the same units are consistently used throughout all the calculations, but in (14) a constant R is dealt

with whose values have been calculated and tabulated, hence it is necessary to employ the same units as those used in calculating R. In English-speaking countries,

P = pressure in pounds per square foot;
$\mathbf{V}$ = volume of one pound of gas, in cubic feet; and
T = absolute temperature in Fahrenheit degrees.

(e) The "constant" in equation (13) may now be interpreted as WR where W represents the number of pounds of gas represented by $V_1, V_2, \ldots V_n$. This applies in all cases in which P stands for pressure in pounds per square foot, V for volume in cubic feet, and T for degrees Fahrenheit above absolute zero.

(f) Boyle's law as previously stated may now be seen to be only a special form of equation (13) or (14). Writing these

$$PV = WRT \quad \text{and} \quad P\mathbf{V} = RT,$$

it becomes evident that the right-hand member is a more complete expression for the "constant" of equation (6).

21. The Specific Heats of Ideal Gases. As a result of the assumption regarding the constitution of ideal gases (see page 29), it follows that, if a quantity of heat, ΔQ, is added to an ideal gas in such manner that it does not alter the gas chemically nor change its motion as a whole, equation (1) must become

$$\Delta Q = \Delta S + \Delta E \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

because, as there are no positive or negative internal forces to overcome in such a gas, *no heat is needed for doing internal work;* hence ΔI equals zero. Therefore the specific heat of an ideal gas, or the heat required when the temperature of an ideal gas is raised one degree, can only supply that needed to increase the temperature and to do the external work corresponding to any resultant volume change with the displacement of surrounding media.

22. Constant-Volume Specific Heat of Ideal Gas (C_v). (a) If the volume of an ideal gas is maintained constant while the temperature is raised, the pressure will increase according to Charles' law. As the volume does not change, no external work can be done because no external media are displaced. All the heat supplied must then be used for changing the temperature;

that is, using ΔQ_v to designate the heat supplied with volume maintained constant,

$$\Delta Q_v \text{ must equal } \Delta S_v.$$

If C_v represents the specific heat of an ideal gas when heated at constant volume, this equation may be written

$$\Delta Q_v = \Delta S_v = WC_v(T_2 - T_1), \quad . \quad . \quad . \quad . \quad (16)$$

from which

$$C_v = \frac{\Delta Q_v}{W(T_2 - T_1)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

W as before standing for the weight of the gas and $(T_2 - T_1)$ being the temperature change. As already explained, the value of C_v thus obtained might be either a real constant or it might be an average over the range from T_1 to T_2.

(b) Obviously this specific heat is a **True Specific Heat** as defined on page 15. Also, it is not only the heat required to raise the temperature of one pound one degree at constant volume, but is likewise the amount given out when the temperature drops one degree under the same conditions.

(c) The symbol C_v denotes a certain quantity of heat energy measured in British thermal units, but as it is sometimes necessary to refer to the same quantity of energy in the mechanical form it is convenient to have a symbol for that purpose. For this K_v is used. As C_v stands for a certain number of thermal units, each of which is equal to 778 foot-pounds (see page 6), it follows that K_v must numerically be 778 times as large as C_v; that is,

$$778\, C_v = K_v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

expresses the relation between the constant-volume specific heat in heat units and in units of mechanical energy.

(d) It is now pertinent to inquire whether C_v is a constant for all conditions of the same gas. That is, whether it takes the same amount of heat to raise the temperature of unit weight one degree at constant volume when the gas is at a high temperature and when it is at a low temperature; whether it takes the same amount of heat with the gas at a low pressure but occupying a large volume as it does with the gas at a high pressure but occupying a small volume.

Experiment and reasoning lead to the belief that C_v may be considered constant for all temperatures and pressures in the

case of the ideal gas, that is, one having only the properties assigned to that material in previous paragraphs.

In the case of real gases, experiments prove C_v to change with variation of temperature and pressure; but for ordinary gases through usual temperature ranges the variations are negligible. With exceptionally high temperatures, such as those occurring in furnaces and the cylinders of internal-combustion engines, the increase in the values of C_v is very noticeable. Therefore it is customary to treat C_v as a constant for real gases except when making accurate calculations for very high temperature conditions.

(e) Since, as shown on page 11, the intrinsic heat energy of a substance depends only upon the content of sensible heat and the heat expended on internal work, it follows from the foregoing that *in the case of ideal gases the intrinsic heat energy depends only on the temperature.* It is impossible to measure the *total* intrinsic heat energy of a gas because it cannot be completely removed. It is, however, possible to measure the quantities concerned in *changes* of intrinsic energy, and this is what is commonly done. Whatever the conditions of the change, if the temperature of W pounds of gas is altered from T_1 to T_2, the **Change of Intrinsic Heat Energy** is

$$\Delta Q_v = WC_v\ (T_2 - T_1). \quad . \quad . \quad . \quad . \quad (19)$$

23. **Constant-Pressure Specific Heat** (C_p). (a) If the pressure of an ideal gas is maintained constant while the temperature is raised, the volume will increase according to Charles' law. As the volume changes, surrounding media must be moved under the constant pressure they exert upon the gas, and hence heat is expended not only in adding intrinsic energy, but also in doing external work ΔE_p. The amount of heat required is, then,

$$\Delta Q_p = \Delta S_p + \Delta E_p = WC_p\,(T_2 - T_1), \quad . \quad . \quad . \quad (20)$$

from which the specific heat at constant pressure C_p is,

$$C_p = \frac{\Delta Q_p}{W\,(T_2 - T_1)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Here, as in the preceding case, the specific heat may also be given in units of mechanical energy, in which case the symbol K_p is used. Evidently

$$778\,C_p = K_p. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

(b) It was shown in Eq. (19) that the change of intrinsic energy depends only on temperature change and is independent of pressure and volume conditions. Hence, whether an ideal gas is heated one degree at constant pressure or at constant volume, the change of *sensible* heat is exactly the same, and it follows that *the constant-pressure specific heat exceeds the constant-volume specific heat by just the quantity of energy necessary to do external work*, ΔE_p. That is, for any temperature variation, T_1 to T_2,

$$\Delta S_p = \Delta S_v,$$

giving $$\Delta Q_p = \Delta Q_v + \Delta E_p,$$

and $$C_p = C_v + \frac{\Delta E_p}{W(T_2 - T_1)}. \quad . \quad . \quad . \quad . \quad (23)$$

(c) A simple expression may be obtained for the external work done when a gas is heated at constant pressure. Imagine, for instance, that one pound of gas with conditions V_1, P_1, T_1, is confined in a cylinder with a movable piston of area F square feet. The length of the portion of the cylinder lying between the inside of the head and the face of piston must be

$$L_1 = \frac{V_1}{F} \text{ feet.}$$

If now the gas is heated at constant pressure to T_2 the volume will increase to V_2 and the piston must move out so that the distance from the inside of the cylinder head to the face of the piston, becomes

$$L_2 = \frac{V_2}{F} \text{ feet.}$$

But the piston will have moved through the distance $L_2 - L_1$ against a force P_1 pounds per square foot; then the work done is

$$\begin{aligned} \text{External work} &= FP_1(L_2 - L_1) \\ &= P_1(FL_2 - FL_1) \\ &= P_1(V_2 - V_1) \text{ ft.-lbs.}; \quad . \quad . \quad . \quad (24) \end{aligned}$$

hence *the external work done when a gas is heated at constant pressure is the product of that constant pressure and the change of volume*, the result being in foot-pounds. If measured in thermal units it would be

$$\text{External work} = \Delta E_p = \frac{P(V_2 - V_1)}{778} \text{ B.t.u.}, \quad . \quad . \quad (25)$$

and if this be substituted in Eq. (23), understanding V_1 and V_2 to stand for the volumes before and after a *one-degree* change, and remembering that the equation must be written in terms of the volume of one pound of gas, there results

$$C_p = C_v + \frac{P(\mathbf{V}_2 - \mathbf{V}_1)}{778}. \quad . \quad . \quad . \quad . \quad . \quad (26)$$

Using the foot-pound symbols

$$K_p = K_v + P(\mathbf{V}_2 - \mathbf{V}_1). \quad . \quad . \quad . \quad . \quad . \quad (27)$$

(d) The constant-pressure specific heat may now be shown to be a constant for an ideal gas. It has been seen that the constant-volume specific heat may be considered constant; then, if $P(\mathbf{V}_2 - \mathbf{V}_1)$ can be shown to be independent of temperature and pressure conditions, K_p and C_p must be constant. This amounts to proving that ΔE_p per degree is constant at all temperatures, pressures, and volumes.

To do this, imagine one pound of a gas with conditions $\mathbf{V}_1$, P_1, T_1, first heated *one degree* to T_1' with volume changing to $\mathbf{V}_1'$ and pressure remaining constant. The external work done will be

$$\text{External work} = P_1(\mathbf{V}_1' - \mathbf{V}_1) \quad . \quad . \quad . \quad . \quad (28)$$

Then, if the condition of the same gas is changed to $\mathbf{V}_2$, P_2, T_2, which may be any values different from $\mathbf{V}_1$, P_1, T_1, and if its temperature is then raised *one degree* to T_2' with the volume changing to $\mathbf{V}_2'$ and the pressure remaining constant, the external work, as before, will be

$$\text{External work} = P_2(\mathbf{V}_2' - \mathbf{V}_2). \quad . \quad . \quad . \quad . \quad (29)$$

But by the law of ideal gases

$$\frac{P_1\mathbf{V}_1}{T_1} = \frac{P_1\mathbf{V}_1'}{T_1'} = \frac{P_2\mathbf{V}_2}{T_2} = \frac{P_2\mathbf{V}_2'}{T_2'} = R.$$

Substituting from this in Eqs. (28) and (29) gives

$$\text{Work in first case} = R(T_1' - T_1) = R,$$
$$\text{Work in second case} = R(T_2' - T_2) = R,$$

so that the work is the same in each case. It follows that *when the temperature of one pound of a perfect gas is raised one degree at constant pressure, the work done is always the same and is independent of the values of* P, $\mathbf{V}$, *and* T; that is, ΔE_p per degree is constant.

More than this, it appears that R, previously known only as the constant in the law of ideal gases, is really equal to the **foot-pounds** of **external work** done when the temperature of *one pound* of gas is raised *one degree* at *constant pressure*. From this

$$K_p = K_v + R, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

and

$$C_p = C_v + \frac{R}{778}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

24. The Ratio γ. The ratio between the two specific heats of gases just considered is of great importance and is designated by the letter γ, thus,

$$\frac{K_p}{K_v} = \frac{C_p}{C_v} = \gamma, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

from which it is evident that γ must always be a quantity greater than unity.

By means of the ratio γ a form of expression, which will be very convenient later, can now be developed from Eq. (30). Rearranging the latter and dividing by K_v, there results

$$1 = \frac{K_p}{K_v} - \frac{R}{K_v} = \gamma - \frac{R}{K_v},$$

from which

$$K_v = \frac{R}{\gamma - 1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

which is the expression sought.

25. Table of Gas Constants. (a) The Gas Constants most commonly used by the engineer are given in Table I. In it all columns with headings having the same subscript are based upon data of the same character. Columns headed with the same letter, as B_1, B_2, B_3, contain values of the same quantity, for each gas, determined in different ways. In the first column of each of the groups B, C, and D is a closely approximate value calculated from data given in column A_1 rounded off to the nearest half-unit. Column A_2 gives molecular weights based on the 1909 International Atomic Weights, and all other columns headed with the subscript 2 contain values computed from these weights. Columns headed with subscript 3 give experimentally determined values.

(b) The calculated columns for Density depend on the **Law of Avogadro.** *This law states that equal volumes of all gases, at the same temperature and pressure, contain an equal number of*

molecules. Hence the densities of different gases must be in the same proportions as the molecular weights. The Weight per Cubic Foot, and Cubic Feet per Pound, columns C and D, can be obtained from the density columns.

(c) While the specific heat C_p would be constant for an ideal gas, it is not constant for real gases. Experiment shows it to vary with temperature and pressure. The values tabulated in column E_3 are *average* values for ordinary temperature ranges at atmospheric pressure.

The values of γ, however, which are usually determined from the velocity of sound in the gas, are generally for some *definite* temperature. Therefore the numbers tabulated in column G_3 are not really on the same temperature basis as those given in column E_3; hence, since the most satisfactory way of obtaining values of C_v is by using the equation $C_v = \frac{C_p}{\gamma}$, the values given in column F_3 must be in error with respect to the rest of the data.

This is well shown by the variation of values for R in column I_3' from those in columns I_2 and I_3. The values in I_3' are obtained from the equation $K_p - K_v = R$, which should give very accurate results and the numerical values show wide variations from those determined in other ways.

In general, it may be said that the amount by which the experimental results vary from the exact calculated values is a measure of the degree of imperfection of the gas under consideration.

(d) For average engineering work it will suffice to use the approximate or observed columns, dropping all but three significant figures; further figures are given in the table to correspond to the supposed standard of accuracy of experimental determinations.

(e) The method of calculating the values in the several columns is as follows:

$$B_1 = 1 \times \frac{A_1}{29} \qquad D_1 = 1/C_1$$

$$D_2 = 1/C_2$$

$$B_2 = 1 \times \frac{A_2}{28.95} \qquad D_3 = 1/C_3$$

B_3 = Observed $\qquad E_3$ = Observed

$C_1 = .080725 \times B_1 \qquad F_3 = E_3/G_3$

$C_2 = .080725 \times B_2 \qquad G_3$ = Observed

$C_3 = .080725 \times B_3 \qquad H$ = Theoretical

TABLE I.—Gas Constants.

Name of Gas.	Molecular Formula.	Atoms per Molecule.	A_1	A_2	B_1	B_2	B_3	C_1	C_2	C_3
			Molecular Weight.		Density, Air=1			Weight in Pounds per Cubic Foot Measured at 32° F. and 29.92 in Hg.		
			Approximate $O_2=32$	International–1909 $O_2=32$	Calculated from Approximate Molecular Weight.	Calculated from International — 1909 Molecular Weight.	Observed.	Calculated from Approximate Molecular Weight.	Calculated from International — 1909 Molecular Weight.	Observed.
Air	.....	.....	29	28.95	1.0000	1.0000	1.0000	0.080725	0.080725	0.080725
Argon	Ar	1	40	39.90	1.3793	1.3782	1.379	0.11135	0.11127	0.11134
Helium	He	1	4	4.00	0.13793	0.13817	0.1368	0.011135	0.011154	0.011043
Carbon Monoxide	CO	2	28	28.00	0.96552	0.96715	0.96716	0.077935	0.078070	0.078071
Chlorine	Cl_2	2	71	70.92	2.4483	2.4498	2.4910	0.19763	0.19776	0.20110
Hydrochloric Acid	HCl	2	36.5	36.468	1.2587	1.2597	1.2692	0.10166	0.10170	0.10246
Hydrogen	H_2	2	2	2.016	0.068965	0.069635	0.06960	0.005565	0.005621_0	0.005618_3
Nitric Oxide	NO	2	30	30.01	1.0345	1.0365	1.0387	0.083510	0.083670	0.083853
Nitrogen	N_2	2	28	28.02	0.96552	0.96786	0.96717	0.077935	0.078130	0.078070
Oxygen	O_2	2	32	32.00	1.1035	1.1054	1.10535	0.089073	0.089227	0.089228
Carbon Dioxide	CO_2	3	44	44.00	1.5172	1.5198	1.5201	0.12247	0.12268	0.12343
Hydrogen Sulphide	H_2S	3	34	34.086	1.1723	1.1775	1.1895	0.094627	0.095047	0.096025
Nitrous Oxide	N_2O	3	44	44.02	1.5172	1.5206	1.5301	0.12247	0.12275	0.12352
Sulphur Dioxide	SO_2	3	64	64.07	2.2070	2.2132	2.2039	0.17815	0.17863	0.18278
Water	H_2O	3	18	18.016	0.62073	0.62234	.....	$0.050I_0$	0.050237	.....
Acetylene	C_2H_2	4	26	26.016	0.89655	0.89863	0.920	0.072375	0.072540	0.074265
Ammonia	NH_3	4	17	17.034	0.58622	0.58842	0.59710	0.047321	0.047500	0.048200
Methane	CH_4	5	16	16.032	0.55175	0.55375	0.55760	0.044537	0.044702	0.045012
Methyl Chloride	CH_3Cl	5	50.5	50.484	1.7413	1.7438	1.7310	0.14057	0.14078	0.13975
Ethylene	C_2H_4	6	28	28.032	0.96552	0.96827	0.98520	0.077935	0.078160	0.079527

TABLE I (*Continued*). — Gas Constants.

Name of Gas.	D_1	D_2	D_3	E_3	F_3	G_3	H	I_3	I_3'	I_3
	Cubic Feet per Pound Measured at 32° F. and 29.92 in Hg.									
	Calculated from Approximate Molecular Weight.	Calculated from International — 1909 Molecular Weight.	Observed				Theoretical			
				Cp	Cv	γ	γ	R	R	R
Air	12.387	12.387	12.387	0.2374	0.16915	1.4037	1.41	53.345	53.12	53.348
Argon	8.0815	8.0876	8.0820	0.1205	0.07259	1.66	1.66	38.708	37.264	38.682
Helium	89.815	89.658	90.560	1.25	0.7622	1.64	1.66	386.12	379.50	390.00
Carbon Monoxide	12.831	12.808	12.811	0.2425	0.17335	1.3989	1.41	55.157	53.78	55.166
Chlorine	5.0600	5.0507	4.0753	0.1241	0.09290	1.336	1.41	21.774	24.28	21.425
Hydrochloric Acid	0.8430	0.8330	0.7602	0.1040	0.13008	1.305	1.41	42.347	42.75	42.033
Hydrogen	170.05	177.00	178.000	3.410	2.4358	1.400	1.41	766.10	758.2	766.50
Nitric Oxide	11.074	11.054	11.025	0.2317	0.1662	1.394	1.41	51.452	50.94	51.357
Nitrogen	12.832	12.800	12.809	0.2438	0.1735	1.405	1.41	55.116	54.69	55.162
Oxygen	11.228	11.208	11.206	0.2175	0.1551	1.4025	1.41	48.200	48.57	48.257
Carbon Dioxide	8.1658	8.1512	8.1018	0.2008	0.1548	1.297	1.28	35.100	35.78	34.892
Hydrogen Sulphide	10.567	10.522	10.414	0.2451	0.1921	1.276	1.28	45.304	41.23	44.850
Nitrous Oxide	8.1658	8.1472	8.0958	0.2126	0.16456	1.292	1.28	35.083	37.37	34.865
Sulphur Dioxide	5.6136	5.5987	5.4710	0.1544	0.1229	1.256	1.28	24.104	24.485	23.562
Water	19.960	19.906	.....	0.4433	0.3444	1.287	1.28	85.715	76.94	.....
Acetylene	13.817	13.786	13.466	(0.35)	.....	1.26	1.21	59.358	.....	57.983
Ammonia	21.133	21.053	20.748	0.5202	0.40108	1.2971	1.21	90.660	92.69	89.360
Methane	22.452	22.373	22.218	0.5930	0.4303	1.3160	1.17	96.335	126.59	95.680
Methyl Chloride	7.1145	7.1035	7.1560	(0.24)	.....	1.2390	1.17	30.591	.....	30.817
Ethylene	12.832	12.7935	12.574	0.4040	0.3173	1.216	1.14	55.092	67.46	54.152

I_2* (Calculated from Inter. Atom. Wts. using Air as a Base.)

$$= \frac{12.387 \times 14.696 \times 144}{491.4 = (273 \times 9/5)} \times \frac{1}{B_2} = R.$$

I_3' (Calculated from Specific Heats.)

$$= 778\ C_p - 778\ C_v = R.$$

I_3* (Calculated from Observed Volumes.)

$$= \frac{D_3 \times 14.696 \times 144}{491.4 = (273 \times 9/5)} = R.$$

* Although the value 492° is used in other parts of this book as the absolute temperature corresponding to 32° F., the value used in computing the quantities given in columns I_2 and I_3 is 491.4, because this corresponds to 273° C., which is used in most experimental determinations.

CHAPTER V.

EXPANSIONS AND COMPRESSIONS OF GASES.

26. Volume Changes. (a) The law of ideal gases expressed by Eq. (13) shows that *there are three inter-related variables which define the condition of a gas; namely, pressure, volume, and temperature. The fixing of any two of these determines the third.* For the study of heat engines it is convenient to consider the behavior of the gas while the volume changes in various ways and to note the accompanying variations of the other two quantities.

(b) Certain diagrams are useful in studying such changes, the most common one being called the **pressure-volume diagram or PV-diagram.** To construct this diagram pressures are plotted vertically and volumes horizontally as in Fig. 6.

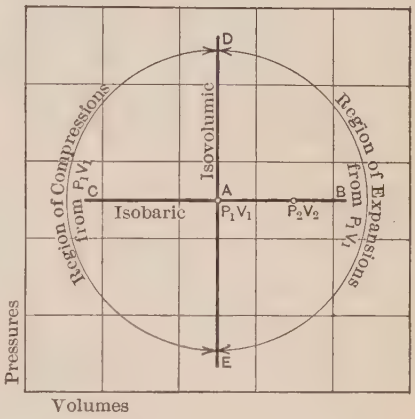


Fig. 6.—PV-Diagram.

Assume, for instance, that the volume and pressure of a given weight of a certain gas are, P_1, V_1, as plotted at A. The volume of this gas may be increased or decreased in different ways. For example, volume might be made to increase or to decrease while pressure is maintained constant. If the various volumes assumed are plotted at constant pressure P_1, the resulting points must lie on the line AB or on the line AC. Either one line or the other would then show graphically the relations of pressure and volume. The temperature at any point as $P_2 V_2$, where $P_2 = P_1$ in this case, could be found by substitution in Eq. (13) or (14), remembering that if the form involving R is used **V** stands for the volume of *one pound* of gas.

(c) Volume-changes with volume *increasing* are known as **expansions**; volume-changes with volume *decreasing* are known as **compressions.** Inspection of Fig. 6 shows that any number of different kinds of expansions are possible; any line drawn from A in the field to the right of DE represents a possible expansion. Similarly, any line drawn from A in the field to the left of DE represents a possible compression. Really, all the expansions commonly used lie in the quadrant between AB and AE and all the usual compressions between AD and AC.

(d) Since there are thus an infinite number of possible methods of changing volume, it is impossible to analyze all of them. Fortunately, the study of **four characteristic methods of change** — including the limiting case with volume constant — suffices for the elementary consideration of heat engines. They are:

1. Volume changes, in which the pressure is constant; or, otherwise stated, in which the volume changes most rapidly with respect to pressure. These are known as **Isobaric Changes** and are represented by the graph AB or AC, Fig. 6.

2. Pressure changes, in which the volume is constant; or, otherwise stated, in which the pressure changes most rapidly with respect to volume. These are known as **Isovolumic Changes** and are represented by the graph AD or AE.

3. Volume changes at constant temperature, known as **Isothermal Changes.**

4. Volume changes known as **Adiabatic Changes** (to be defined later).

27. Constant-Pressure or Isobaric Changes of Gases. As just explained, the graph of such changes is a horizontal line on the PV-diagram. In Fig. 7 an isobaric expansion of gas with initial conditions $P_1 V_1$ would be represented by the line AB and a similar compression by the line AC.

Equation for Isobaric Changes.

(a) It is evident from an inspection of the graph of such a change, or from the definition, that its equation in PV coördinates is

$$P = \text{Constant.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Change of Associated Heat during Isobaric Changes of Gases.

(b) During these volume changes at constant pressure the temperature must vary according to Charles' law; that is, the absolute temperature must vary directly as the volume. But when a unit weight of gas has its temperature raised or lowered one degree at constant pressure, it absorbs, or gives out, a quantity of heat equal to its constant-pressure specific heat, C_p. Then for any weight W of gas changing volume from V_1 to V_2 at constant pressure with a corresponding change of temperature from T_1 to T_2, the *change of associated heat* (in thermal units) is

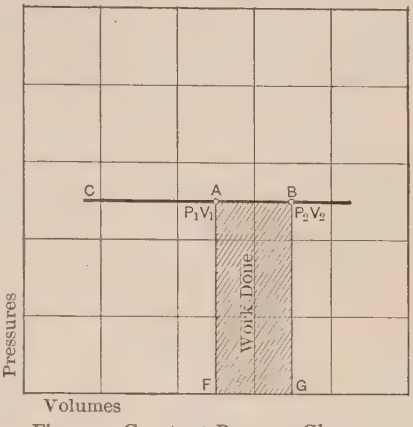


Fig. 7.—Constant-Pressure Changes

$$\Delta Q = WC_p(T_2 - T_1). \quad . \quad . \quad . \quad . \quad . \quad (35)$$

If the volume change is an expansion the result will be positive because T_2 will be greater than T_1; but if the volume change is a compression T_2 will be less than T_1 and the result will be negative. Negative heat change must be interpreted as heat given out or liberated, so that the equation as stated is true for compressions as well as expansions.

Work during Isobaric Changes.

(c) It was shown in Eq. (24) that the external work in foot-pounds done by a gas when its temperature increases with pressure constant is given by the equation

$$\text{External Work} = 778\,\Delta E = P(V_2 - V_1)\ \text{ft.-lbs.} \quad . \quad . \quad (36)$$

This is the equation for work done during a constant-pressure volume change. The equation gives a positive result for expansion, and a negative one for compression; that is, a gas *expanding* at constant pressure *does work*, and *work must be done to compress* a gas at constant pressure.

(d) Obviously, the product $P \cdot (V_2 - V_1)$ is represented in Fig. 7, by the crosshatched area under the line showing the volume change from V_1 to V_2, hence the area under that line, expressed in foot-pounds, is equal to the work done during the volume change. For an expansion the area is interpreted as positive, that is, as the area $ABGF$; for a compression from $P_2\ V_2$ to $P_1\ V_1$, as negative, that is, the area $BAFG$. This property of representing work by area is common to all PV-diagrams and makes them very useful.

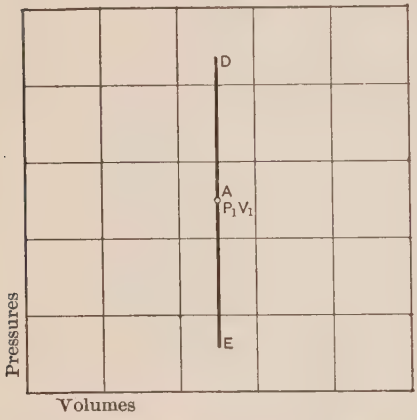


Fig. 8. — Constant-Volume Changes

28. Constant-Volume or Isovolumic Changes of Gases. (a) With volume constant the pressure will increase as the absolute temperature increases, and decrease as that temperature decreases. The graph of such changes plotted to PV coördinates is a vertical line like ED in Fig. 8. There being no volume change, there can be no expansion or compression. A line drawn vertically upward, as AD, means pressure increase, and a line drawn vertically downward means pressure decrease.

Equation for Isovolumic Changes.

(b) The equation of constant-volume changes in terms of P and V must be

$$V = \text{Constant}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

Changes of Associated Heat during Isovolumics of Gases.

(c) To increase the temperature of one pound of gas one degree at constant volume requires a heat addition equal to C_v thermal units, and to decrease the temperature one degree an amount of heat equal to C_v must be withdrawn. Then if the pressure of W pounds of gas changes from P_1 to P_2 at constant volume, while the temperature varies according to Charles' law

from T_1 to T_2, the change of associated heat (in thermal units) is

$$\Delta Q = WC_v (T_2 - T_1). \quad . \quad . \quad . \quad . \quad . \quad (38)$$

For a pressure increase Charles' law shows that T_2 must be greater than T_1 and the result will be positive. For a pressure decrease, T_2 will be less than T_1 and the result will be negative; that is, heat must be supplied to cause increase of pressure at constant volume, and must be abstracted to cause decrease of pressure at constant volume.

Work during Isovolumic Changes.

(d) Since there is no change in volume, — that is, no displacing of surrounding media, — there can be no external work done. Then for this case, in foot-pounds,

$$\text{External Work} = 778\,\Delta E = O. \quad . \quad . \quad . \quad (39)$$

In the figure given there is no area under the line representing the change, and therefore, since area on the pressure-volume diagram represents work, it follows that the work equals zero.

29. Constant-Temperature or Isothermal Changes of Gases. (a) If the temperature of a gas is maintained constant, while its pressure and volume change, Boyle's law applies and gives the relation between these two variables as

$$PV = \text{Constant}.$$

If, starting with $P_1 V_1$ in Fig. 9, different values be substituted for V in this equation and the resulting pressures are computed and plotted against these volumes, the graph obtained will be CB, which is a rectangular hyperbola. The line from A to B shows an isothermal expansion from $P_1 V_1$ and the line from A to C represents an isothermal compression from the same point.

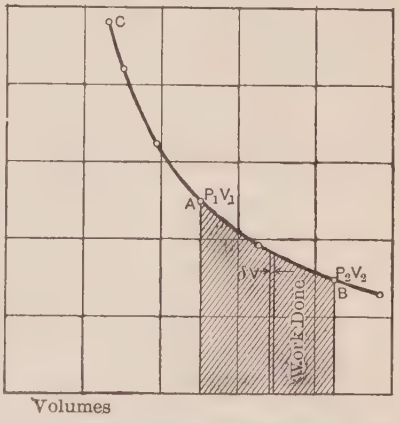


Fig. 9. — Isothermal Changes for Ideal Gases

Equation for Isothermal Changes of Gases.

(b) The equation of these changes, in terms of pressure and volume, must be that just given,

$$PV = \text{Constant}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

Work during Isothermal Changes of Gases.

(c) Since there is a change of volume during isothermal changes, external work must be done. If in Fig. 9 the two closely spaced vertical lines represent a volume change δV, from V_1' to V_2', so small that the pressure may be assumed constant while it is taking place, the external work in foot-pounds during that small change must be

$$778\ \delta E = P(V_2' - V_1') = P[(V_1' + \delta V) - V_1'] = P\,\delta V$$

and for a finite change of any size, from V_1 to V_2

$$778\ \Delta E = \int_{V_1}^{V_2} P\,dV. \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

To integrate this expression, it is necessary to substitute for P in terms of V. Assuming P and V as the values for any point on the curve and P_1 and V_1 as the values with which the expansion starts,

$$P_1V_1 = PV$$

from which

$$P = \frac{P_1V_1}{V}$$

Substituting this value in the expression (41) gives

$$\begin{aligned} 778\ \Delta E &= \int_{V_1}^{V_2} \frac{P_1V_1}{V}\,dV = P_1V_1 \int_{V_1}^{V_2} \frac{dV}{V} \\ &= P_1V_1(\log_e V_2 - \log_e V_1) \\ &= P_1V_1 \log_e{}^{*}\frac{V_2}{V_1} \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad (42) \end{aligned}$$

If the ratio of volumes (V_2/V_1) in the last expression, known as the **ratio of expansion** or of compression, is designated by the

* It is usually more convenient to use $\log_{10}$ instead of $\log_e$. Since $\log_e = 2.302 \log_{10}$, Eq. (42) may be written $= P_1V_1 \times 2.302 \log_{10} \frac{V_2}{V_1}$. The other logarithmic equations which are to follow may be similarly transformed.

letter r, the equation for work done by *any weight* of gas may be written

$$778\ \Delta E = P_1 V_1 \log_e r \text{ ft.-lbs.} \quad . \quad . \quad (43a)$$

For *unit weight*

$$778\ \Delta E = RT_1 \log_e r \text{ ft.-lbs.} \quad . \quad . \quad . \quad (43b)$$

If the expansion is negative, that is, if there is compression, the work done by the gas must be negative; thus work must be done upon it to decrease its volume.

It should be noted that $\int_{V_1}^{V_2} P\,dV$ is the general expression for the area under a curve drawn to PV coördinates and hence in Fig. 9 the crosshatched *area on the diagram is a measure of work done.*

Change of Associated Heat during Isothermal Changes of Gases.

(d) In order that a gas may expand and do work, an amount of energy equivalent to the work done must be supplied from some internal or external source. The only heat energy associated with an ideal gas* is that associated with it as *sensible heat*, and that *stored in surrounding media* as a result of previous expansion to the present volume.

In *isothermal* expansion temperature is constant by definition, and since under this condition the internal energy of an ideal gas is constant, it follows that there can be no change in the store of sensible heat of the expanding gas, and therefore that this store cannot be the source of energy for the doing of work.

In *any expansion* the energy stored in external media is *increased*, and therefore this store cannot be a source of energy to do work, during the expansion.

Hence a gas doing work can expand isothermally only if it receives from an external source an amount of energy equal to the work done, and this energy can only be received as heat. It follows, then, that *an ideal gas expanding isothermally and doing work receives from surrounding media during the expansion a quantity of heat equal to the external work done*, hence, from Eq. (43a), for *any weight* of gas,

$$\Delta Q = \Delta E = \frac{P_1 V_1 \log_e r}{778} \text{ B.t.u.} \quad . \quad . \quad (44a)$$

* It is assumed that no chemical change occurs nor any change of motion of the gas as a whole.

And for *unit weight*,

$$\Delta Q = \frac{RT_1 \log_e r}{778} \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad . \quad (44b)$$

Thus during isothermal expansion there is no change of heat *in* the gas itself; the gas merely serves as a conveyor of the added heat ΔQ; and this heat may be considered as external work ΔE, in the solution of problems. Hence, during isothermal expansion in a piston engine the work delivered through the piston rod is equal to this ΔQ.

(e) If a gas is forced to expand, and to do external work, without a supply of heat energy from some external source, it derives the necessary quantity from its own internal store of sensible heat; but this is accompanied by a temperature drop and the expansion cannot be isothermal.

Isothermal *compression* is the reverse of isothermal expansion and the work done on the gas appears as heat, which must be removed as fast as it is generated; otherwise the gas will absorb this energy as sensible heat, with rise in temperature, and the operation cannot then be isothermal.

It may seem at first sight as if isothermal expansion of a perfect gas furnished an exception to the second law of thermodynamics. It is certainly true that all the heat energy supplied to the gas under such conditions is completely converted into mechanical energy, but it is equally true that this process cannot continue indefinitely. That is, such a transformation cannot go on continuously, and must stop in general when the pressure of the expanding material has reached that of the surrounding media.

30. Adiabatic Volume Changes of Gases. (a) During adiabatic expansion or compression *no energy*, **in the form of heat,** *is supplied to, or withdrawn from, the expanding gas*, as would be the case if the walls of the vessel surrounding the gas were of material which is perfectly nonconducting as regards heat. Therefore all heat that is transformed into external work by such expansion must come from the sensible heat store of the working gas, and all work that is transformed into heat by such compression goes to increase the store of sensible heat of the gas.

More briefly — the external work done by the adiabatic expansion of a gas has its energy source in the *sensible heat* of the

gas. The heat resulting from work done in adiabatic compression is stored as sensible heat in the gas.

To illustrate adiabatic changes, imagine a gas confined in a cylinder permanently closed at one end and supplied with a frictionless piston. Assume that the apparatus is all made of material that will neither absorb nor transmit heat. If the piston moves out, the volume of the gas will increase adiabatically and external work will be done; if the piston moves in, the volume will decrease adiabatically and work will be done upon the gas.

(b) During adiabatic volume increase against resistance—that is, during *adiabatic expansion* with the doing of external work — the *temperature* of the gas *must fall* because external work is done at the expense of sensible heat. During adiabatic volume decrease — *adiabatic compression — the temperature must rise* because the heat equivalent of the work of compression goes to increase the sensible heat of the gas. Obviously, in this imaginary operation, the heat that disappears during expansion equals the external work done; and during compression the work of compression equals the heat increase in the gas.

Equation for Adiabatic Changes of Gases.

(c) The equation representing the adiabatic change of a gas has the form

$$PV^n = \text{Const.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45a)$$

which may be rewritten

$$PV^n = P_1V_1^n = P_2V_2^n \; . \; . \; P_xV_x^n = \text{Const.} \quad . \quad . \quad (45b)$$

in which $n = \gamma$, as will be shown when Eq. (48) is derived.

Work during Adiabatic Changes of Gases.

(d) Using reasoning similar to that which led to Eq. (41), the expression for work done during an adiabatic change must be

$$778\ \Delta E = \int_{V_1}^{V_2} P dV.$$

Substituting in this $P = \dfrac{P_1V_1^n}{V^n}$, obtained from Eq. (45), gives

$$778\ \Delta E = \int_{V_1}^{V_2} P_1V_1^n \frac{dV}{V^n} = P_1V_1^n \int_{V_1}^{V_2} \frac{dV}{V^n}$$

$$= \frac{P_1V_1^n\,(V_2^{1-n} - V_1^{1-n})}{(1-n)} \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (46)$$

This can be simplified by performing the multiplication indicated in the numerator, then substituting from the relation

$$P_1V_1^n = P_2V_2^n$$

and cancelling, thus obtaining for *any weight* of gas,

$$778\,\Delta E = \frac{P_1V_1 - P_2V_2}{(n-1)} \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (47a)$$

and for *one pound* of gas,

$$778\,\Delta E = \frac{R\,(T_1 - T_2)}{(n-1)} \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (47b)$$

These equations cannot be used numerically, however, until the value of n is known. This will now be determined.

(e) Since the sensible heat lost by a gas when expanding adiabatically and doing external work must equal the work done, and since, in any case, the sensible heat energy lost per pound of a gas which is changing temperature in any way from T_1 to a lower value T_2, must equal $K_v\,(T_1 - T_2)$ ft.-lbs., — it follows that

$$K_v\,(T_1 - T_2) = \frac{R\,(T_1 - T_2)}{(n-1)}$$

and substituting for K_v its value from Eq. (33)

$$\frac{R\,(T_1 - T_2)}{(\gamma - 1)} = \frac{R\,(T_1 - T_2)}{(n-1)}$$

from which it follows that

$$n = \gamma. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

Then the equation of an adiabatic change is, as was mentioned in connection with Eq. (45),

$$PV^\gamma = \text{Constant.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

(f) The work done is, from Eq. (47a), for *any weight* of gas,

$$778\,\Delta E = \frac{P_1V_1 - P_2V_2}{(\gamma - 1)} \text{ (ft.-lbs.)} \quad . \quad . \quad . \quad (50a)$$

and for *one pound* is, from Eq. (47b),

$$778\,\Delta E = \frac{R\,(T_1 - T_2)}{(\gamma - 1)} \text{ (ft.-lbs.)} \quad . \quad . \quad . \quad (50b)$$

Temperature Change of Gas during Adiabatics.

(g) Since during an adiabatic process the stock of sensible heat, and hence also the temperature, is constantly changing, — dropping during an expansion and rising during a compression, — it is necessary to find some means of determining the extent of this temperature variation. If a gas changes adiabatically from P_1V_1 to P_2V_2, Eq. (49) gives

$$P_1V_1^{\gamma} = P_2V_2^{\gamma}; \quad \text{or} \quad \frac{P_2V_2^{\gamma}}{P_1V_1^{\gamma}} = 1 \quad . \quad . \quad . \quad . \quad \text{(a)}$$

and the law of ideal gases gives

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}; \quad \text{or} \quad \frac{P_1V_1}{P_2V_2} = \frac{T_1}{T_2}. \quad . \quad . \quad . \quad \text{(b)}$$

If the last forms of expressions (a) and (b) be multiplied together, there results

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = r^{\gamma-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

and substitution for $\frac{V_2}{V_1}$ from the first form of (a) gives

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

Either Eq. (51) or (52) can be used for finding the temperature resulting from an adiabatic change if the initial temperature is known.

31. General Expression for Volume Changes. (a) All the common volume changes of gases can be represented with necessary accuracy by the general form of expression,

$$PV^n = \text{Constant.}$$

It is of course assumed that n will have a special numerical value for each different type of change. The truth of this proposition for the changes so far developed can be seen by writing the equations in the following fashion:

For pressure const., P = Const. may be written PV^0 = Const.
For volume const., V = Const. may be written PV^{∞} = Const.
For isothermal, . PV = Const. may be written PV^1 = Const.
For adiabatic, . PV^{γ} = Const. is . . . . PV^{γ} = Const.

(b) **A comparison of the expansion curves** (Fig. 10) will show that *as the graph of the different expansions considered swings down through the quadrant BAE the exponent increases in size. The facts that any equation with* $n < 1 > 0$ *gives a graph less steep than the isothermal and that any equation with* $n > 1 < \infty$ *gives one steeper than the isothermal* are very useful and should be memorized. The idea of *steepness* in this statement is important, as in general one curve cannot be said to lie above or below another. For example, if the curves in Fig. 10 are continued as compressions, to the left from the point A, their relations as to vertical position are reversed.

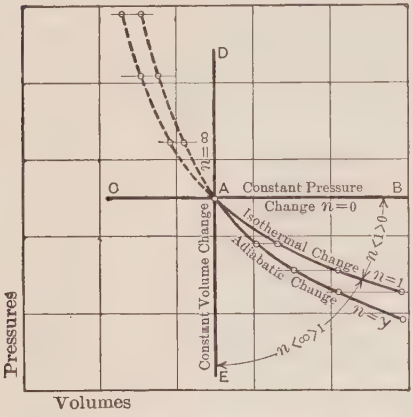


Fig. 10. — Showing Effect of Value of n in Equation PV^n = Const.

32. Construction of Lines Representing Volume Changes. (a) In dealing with heat engines it is frequently necessary to construct the lines representing graphically the changes already discussed and others of similar character. This can always be done by substituting assumed volumes or assumed pressures in the equation for the type under consideration, solving for the other quantity and plotting the resulting points. The curve joining these points is the graph sought. An exponential form of equation involves the use of logarithms and in some cases the calculations become a little more troublesome.

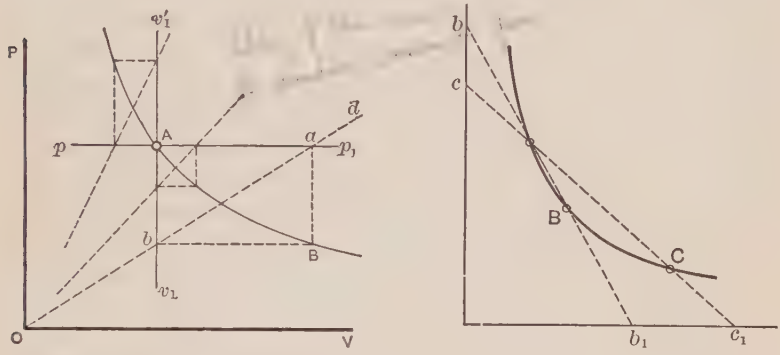
It is therefore convenient to know graphical methods of determining directly the curves representing the various changes.

Graphical Construction of Curve PV = Constant.

(b) In Fig. 11, with coördinates P and V as before, let it be desired to draw an equilateral hyperbola through point A. For doing this two methods will be given. **First Method**: — Draw through the point A horizontal and vertical lines pp_1 and v_1v_1'; next, from the origin O draw any number of rays (such as Od)

to intersect these lines (as at a and b); then horizontal and vertical lines drawn through these points of intersection will meet at points (such as B) on the desired curve. For expansion from point A, the rays fall below A; for compression, they fall above.

Second Method: — Through A, Fig. 12, draw any number of



Figs. 11 and 12.— Construction of Curve PV = Constant on PV-Chart.

lines, as bb_1, cc_1, etc.; make $b_1B = Ab$, $c_1C = Ac$ and so on; then the points A, B, C, etc., will be on the desired curve.

Construction of Curve PV^n = Const. by Using Logarithmic Cross-Section Paper.

(c) The equation $PV^n = K = const.$, if solved by logarithms, takes the form $\log P = -n \log V + \log K$. Then letting $y = \log P$, $x = \log V$, and $k = \log K$, the last equation may be rewritten $y = -nx + k$. This is the equation of a straight line with negative slope n and y-intercept k. It is shown in Fig. 13 by KS, drawn to the ordinary *uniform scales* O_X and O_Y. The abscissa of any point A on KS, measured on the scale O_X, gives the logarithm of the value of V represented; its ordinate on scale O_Y is the logarithm of the value of P.

If now new scales L_v and L_p are constructed in such a manner that the lengths 1–2 and 1–3, etc., represent the logarithms of 2, 3, etc., — as is done on the scale of the slide rule, — then these **logarithmic scales** may be used for reading *directly* the numerical values of P and V corresponding to points on KS. To the scale L_p, the y-intercept of this line is the constant K, and to the *uniform* scale the slope is n. Since the values of P and V

may be read directly on the logarithmic scales, the uniform scales are not usually given on charts of this character.

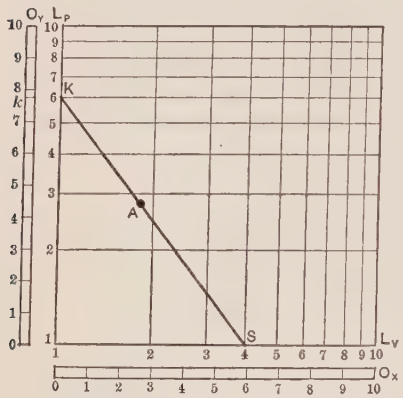


Fig. 13.—Logarithmic Chart.

Any straight line can be located on the logarithmic chart if the PV values for two points are known; or if one point and the slope n are given; but in the latter case it must be remembered that the slope is laid off using the uniform scale. For one of the points it is sometimes convenient to use the y-intercept, K.

After the line has been drawn the simultaneous values of P and V may be read, and these may then be used in drawing the PV curve on ordinary cross-section paper.

The chart in Fig. 13 is arranged for numbers between 1 and 10, but it may be used for numbers between .1 and 1, between 10 and 100, 100 and 1000, and so on, by merely changing the scales to suit.

When a wider range of numbers is under consideration (as from .1 to 100) a "checkerboard" composed of several similar logarithmic charts may be used. Thus in Fig. 14, let each of the squares contain such a chart and let the one surrounded by heavy lines correspond to Fig. 13, with KS reproduced. In the lower tier of charts the ordinates are for numbers from .1 to 1, in the middle tier they are from 1 to 10, and in the upper from 10 to 100. The abscissas for the vertical columns progress, from left to right, by multiples of ten also.

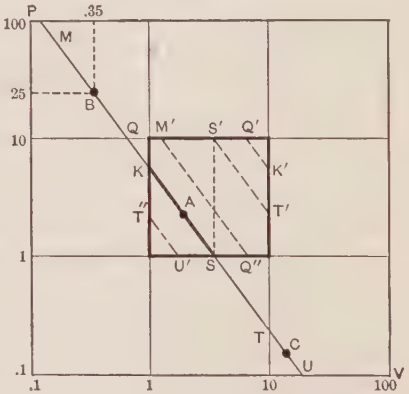


Fig. 14.—Checkerboard of Logarithmic Charts.

The coördinates of a point anywhere on the checkerboard can

be read directly on the proper scales. For example, point B has coördinates $P = 25$ and $V = .35$; for point C the value of P is .14 and V is 13.

As all the squares are cross-ruled the same, and differ only as to scales (and that by multiples of 10), it is evident that if $S'T'$ is drawn in the central chart in a position similar to that of ST in the square below, it can be used in place of the latter line provided points on it are read to a scale $\frac{1}{10}$ that used for KS. Similarly, $T''U'$, $K'Q'$, and $Q''M'$ may be drawn to correspond to TU, KQ, and QM respectively, and may be used instead of them with proper change in scale. Thus a single logarithmic chart may be used in place of the checkerboard. Obviously, when the curve crosses a horizontal boundary line the scale of ordinates changes; and when it crosses a vertical boundary the scale of abscissas changes.

If $n = 1$, as in the case of the equilateral hyperbola, the slope of the line is $-45°$. If the exponent is greater than 1, the slope is steeper, and *vice versa*.

TABLE II. — Collected P, V, T Formulas for Volume Changes of Gases.†

Kind of Change	Equation for	Final Pres. $P_2 =$	Final Volume	Final Temp. $T_2 =$	B. t. u. given to Unit Wt.*	Ft.-lbs. Work by Unit Wt.*
Isobaric or Constant Pressure	$P = Const.$	P_1 $R\frac{T_2}{\mathbf{V}_2}$	$V_2 = V_1\frac{T_2}{T_1}$ $\mathbf{V}_2 = R\frac{T_2}{P_2}$	$T_1\frac{V_2}{V_1}$ $\frac{P_2\mathbf{V}_2}{R}$	$+ C_p(T_2 - T_1)$	$+ P_1(\mathbf{V}_2 - \mathbf{V}_1)$
Isovolumic or Constant Volume	$V = Const.$	$P_1\frac{T_2}{T_1}$ $R\frac{T_2}{\mathbf{V}_2}$	$V_2 = V_1$ $\mathbf{V}_2 = R\frac{T_2}{P_2}$	$T_1\frac{P_2}{P_1}$ $\frac{P_2\mathbf{V}_2}{R}$	$+ C_v(T_2 - T_1)$	o
Isothermal or Constant Temp.	$PV = Const.$	$P_1\frac{V_1}{V_2}$ $R\frac{T_2}{\mathbf{V}_2}$	$V_2 = V_1\frac{P_1}{P_2}$ $\mathbf{V}_2 = R\frac{T_2}{P_2}$	T_1	$+\frac{P_1\mathbf{V}_1\log_e r}{778}$ $+\frac{RT_1\log_e r}{778}$	$+ P_1\mathbf{V}_1\log_e r$ $+ RT_1\log_e r$
Adiabatic	$PV^\gamma = Const.$	$P_1\left(\frac{V_1}{V_2}\right)^\gamma$ $R\frac{T_2}{\mathbf{V}_2}$	$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}}$ $\mathbf{V}_2 = R\frac{T_2}{P_2}$	$T_1\left(\frac{V_1}{V_2}\right)^{\gamma-1}$ $T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$	o	$+\frac{P_1\mathbf{V}_1 - P_2\mathbf{V}_2}{\gamma - 1}$ $+\frac{R(T_1 - T_2)}{\gamma - 1}$
Any change given by eq. in next column	$PV^n = Const.$	$P_1\left(\frac{V_1}{V_2}\right)^n$ $R\frac{T_2}{\mathbf{V}_2}$	$V_2 = V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$ $\mathbf{V}_2 = R\frac{T_2}{P_2}$	$T_1\left(\frac{V_1}{V_2}\right)^{n-1}$ $T_1\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$	Any value depending on variations imposed	$+\frac{P_1\mathbf{V}_1 - P_2\mathbf{V}_2}{n - 1}$ $+\frac{R(T_1 - T_2)}{n - 1}$

* Written for expansions.

† All formulas involving R and $\mathbf{V}$ are for unit weight of gas.

CHAPTER VI.

REVERSIBILITY.*

33. Definition. (a) Processes, or series of changes, which may be made to occur with materials and their associated energies are broadly divided into two kinds:

1. Irreversible processes, and
2. Reversible processes.

An **Irreversible Process** is one which affects the participating materials and energies in such a way that after its total or partial completion it would be impossible to return those materials and energies to initial conditions, without leaving changes in other materials and their associated energies. All the actual processes with which the engineer has to deal are of this character. It is, however, possible to imagine some of these processes as taking place under ideal conditions in such a way that after their completion everything can be again returned to starting conditions without leaving changes in anything, even though it be entirely extraneous to the system under investigation. Such ideal changes are called **Reversible Processes.**

(b) A good mechanical example of a reversible change is furnished by a pendulum swinging on a frictionless support and in a perfect vacuum. Each cycle of the pendulum is accompanied by a change of kinetic to potential energy, and then a reversal of this process so as to bring everything concerned to exactly the conditions pertaining at the start. This, then, is a process which may be said to be by nature reversible.

A real pendulum can never reproduce this ideal process exactly, because of friction at the support and more or less friction in the enveloping medium. These resistances change some of the kinetic energy of the pendulum into heat which in the usual case leaves the system by radiation. Thus, the end of each cycle finds the pendulum system poorer in energy by the amount

* The study of this chapter may be deferred until Section 49 (h) is reached.

of heat which has been lost; and surrounding materials must of course have gained a corresponding amount of energy. The real process does not therefore fulfill the requirements of a reversible process.*

(c) It will be observed from the preceding paragraph that the ideally reversible process becomes imperfectly reversible as soon as losses are assumed to occur. Although the process could never be performed in reality without such losses, this does not invalidate the determination of the laws of the ideal pendulum, — laws which are very valuable for investigations of a certain type. Since a reduction of losses in a real process of this character will cause the process to approach the ideal reversible one more closely, it is evident that the reversible process and the laws derived for it may be regarded as the ideal limiting case of the real process and of the laws governing it. This applies to processes of a certain character only.

(d) There are many other processes of such character that no assumptions of ideal mechanisms and no reasonable assumptions as to the elimination of losses can reduce them to limiting reversible processes, as was done in the case of the pendulum. Such processes are the irreversible ones, examples of which will be given later.

(e) In the investigation of certain thermodynamic transformations accompanying pressure, volume, and temperature changes, it is often possible and desirable to assume all losses absent so that the process may be considered reversible. The assumptions as to the elimination of loss must be *reasonable* ones, however, — those not involving changes in the intrinsic character of the process. Thus it is permissible to assume that there is an absence of friction, and that one may use a material which does not conduct heat, as was done in previous chapters; but an assumption that there is no internal heat energy lost when a gas does work by expanding adiabatically behind a piston would have been unreasonable, as it would have been absolutely contrary to the intrinsic character of the process.

* The process might still be considered reversible if there was any way of gathering up the energy lost as heat, converting all of it into mechanical form, and returning this to the pendulum. But, even assuming that the heat could all be caught, the second law of thermodynamics states that it cannot all be again converted into the mechanical form, and the statement made above must therefore be correct.

The study of an ideal reversible process in lieu of a real imperfectly reversible one greatly simplifies problems and makes possible the development of laws which would otherwise be obtainable only with great difficulty.

(f) For thermodynamic purposes a reversible process may be defined as follows:

A **thermodynamically reversible process** is one involving heat and mechanical energy transformations which are of such character that, after completion, they can be carried through in the opposite sense, without resulting in any changes in anything extraneous to the system under consideration.

The practical application of this simple definition is often confusing. There are certain processes which are obviously reversible in this sense and certain others which are as obviously irreversible; but there are also many about which a decision is difficult. A few reversible and irreversible thermodynamic processes are given in the succeeding sections.

34. Some Reversible Processes. (a) A good example of an ideal thermodynamically reversible process is as follows: Imagine a perfect gas inclosed in a cylinder made of material that will neither absorb nor conduct heat and let it be fitted with a frictionless piston of the same material. If the gas expands it must do so adiabatically, since the heat insulation is assumed to be perfect. The temperature will drop, the volume will increase, and work will be done in driving the piston outward against whatever resistance is offered — for instance, the raising of a weight.

If, after the piston has reached a certain point, the work which has been done by the gas is returned to drive the piston back to its original position — for instance, by the dropping of the lifted weight, — the gas will be compressed adiabatically to its original condition and, in the ideal case, nothing in the universe need have been changed by the process.

Such a process is thermodynamically reversible. It is evidently ideal and can only be approximated in real cases, for every material known absorbs and conducts heat, and no piston can be frictionless.*

* It is really necessary to further stipulate that the expansion and compression of the gas in this process take place at infinitely slow rates, to make it perfectly reversible in theory. This is necessary in order to exclude any degree of what is termed " free expansion," an irreversible phenomenon which will be treated in a later paragraph.

(b) Again, imagine a body which is at a certain temperature, and is so arranged that the withdrawal of heat from it does not change its temperature. The steam in a boiler is a body of material approximating this conception. If a confined body of gas is kept in contact with this source of heat, or hot body,— as in a steam-jacketed cylinder made of perfectly conducting material,— and if it is allowed to expand and do external work, such as driving out a piston against resistance, the expansion must be isothermal. During the process heat will be drawn from the hot body and appear as mechanical energy to do external work. This work may be returned by compressing the gas isothermally and restoring the resulting heat to its original source. The ideal process is thermodynamically reversible, but practically some heat must have been radiated and some lost as friction, so that the reversibility is, as before, ideal only.

(c) A reconsideration of the simple ideal expansions discussed in Chapter V will now show that all of these may be made reversible processes.

35. Some Irreversible Processes. (a) One of the best examples of an intrinsically irreversible process is furnished by the passage of heat from one body to another which is at a lower temperature. Consider two bodies at different temperatures brought into contact and thermally isolated from the rest of the universe. Experience shows that the colder body will receive heat from that having the higher temperature, and that this process will continue until the two temperatures become the same; also it shows that certain physical changes will accompany this passage of heat. Thus there may be a change of state, as would be the case, for instance, if the colder body is ice at the melting point; or again, there may be simply changes in volume accompanying the doing of external work.

No matter what the conditions, no method has yet been devised to reverse this process thermodynamically; that is, to make heat flow from the previously cool body to the other so as to leave them at different temperatures, return them to the initial physical conditions, and leave no change in anything else. The process is then irreversible by definition.

(b) Another example of an intrinsically irreversible process is the free expansion of a perfect gas similar to that which occurs

in Joule's experiment.* Imagine two vessels of equal size joined by a pipe containing a valve, all made of non-heat-conducting material. Imagine further that one vessel contains a quantity of perfect gas at some given pressure and temperature and that the other vessel is absolutely empty. If the valve in the connecting pipe is opened, the gas will rush from the high-pressure vessel into the other one and ultimately both will contain the same quantity of gas at the same pressure and temperature. To make this possible, the gas originally contained in one vessel must have expanded until its volume became sufficient to fill the two. Since the volume occupied by the gas is now greater than before, the pressure must be lower unless the temperature has risen, and it will be found that this has not occurred. Further, since the vessels and connecting pipe are nonconducting, and since the system is so arranged that no disturbance of surrounding media can be caused, it follows that there can have been no loss of heat energy by the gas.

The heat energy associated with the system must therefore be the same before and after the change. Since, however, as was shown in connection with the specific heat of gases, the intrinsic heat energy of a perfect gas is always the same at the same temperature, it follows that *the temperature of the gas must be the same when filling two vessels as when filling one.*

To make this process reversible, it must be possible to compress the gas again into one vessel, keep the temperature the same, and have no change in anything outside the system of two vessels and contained gas. This is impossible, as work would have to be done upon the gas to compress it, and there would then either be a rise in temperature or the heat of compression would have to be absorbed by some body outside the system. This heat, though equal to the work of compression, could not be returned to the engine, or device doing that work, as an equivalent of the work done. This is so because (according to the Second Law of Thermodynamics) no engine could deliver in mechanical form all the heat supplied it.

Obviously the process is intrinsically irreversible because it is impossible to imagine its thermodynamic reversal even with ideal mechanism.

* This is not to be confused with Joule's experiment for the determination of the mechanical equivalent of heat.

(c) The process of free expansion is one of the most interesting and is worthy of more detailed study. What really happens is best considered in two parts.

The gas in the high-pressure vessel begins to expand as soon as the valve in the connecting pipe is opened, and it acquires a high velocity with rapid drop of pressure as it flows into the empty vessel. The kinetic energy associated with this velocity must come from the intrinsic heat energy possessed by the gas. The expansion is in reality adiabatic, and the intrinsic heat energy, which during expansion behind a piston would have been converted into mechanical work, is here converted into the kinetic energy acquired by the gas itself. The temperature of the gas drops just as in the other adiabatic expansions already considered; thus, the material entering the empty vessel has the low temperature which corresponds to the low pressure and is deducible directly from the law of adiabatic expansion.

Considering next the receiving vessel, — the gas with low pressure and temperature enters with high velocity but immediately becomes churned up, impinges on the walls, etc., and slowly comes to rest. The energy originally possessed by virtue of its velocity cannot be destroyed, but is reconverted into heat. It is absorbed as sensible heat by the gas and raises its temperature. Given sufficient time, equilibrium will be established between the material in the two vessels, and then the gas, with the same stock of heat as before, will have returned to the temperature it had initially.*

* The importance of the footnote on page 61 can now be appreciated. If gas, expanding adiabatically behind a piston, is allowed to acquire an appreciable velocity, some of the heat energy which has previously been assumed as doing external work will be used to impart this velocity to the gaseous molecules. In just so far as this occurs the process will be irreversible. In all real piston engines the velocity acquired is so small that the heat thus used is negligible. Hence no attention is ever given the phenomenon from this viewpoint, and it need not be considered in discussing the elementary cycles in the following chapters.

CHAPTER VII.

ENTROPY.

36. Explanatory. In the more advanced discussions of thermodynamic theory a certain property of substances, know as their "*Entropy*" (represented by ϕ), is found to be of great importance. The solution of most, if not all, engineering problems involving thermodynamic changes can be obtained without employing entropy; but its use enables scientists to draw certain sweeping conclusions with regard to natural phenomena,—conclusions which would otherwise be difficult to formulate, and which materially assist in developing the laws governing thermodynamic transformations. The consideration of entropy also serves the useful purpose of giving the engineer a broader viewpoint with regard to the processes he makes use of. For these reasons it is introduced here.

37. Definition. (**a**) It has been seen that it is impossible to measure the absolute amount of associated heat energy (Q), and that all cases can be analyzed when the discussion is limited to *changes* of energy (dQ, δQ, or ΔQ). Later it will be shown that entropy is a similar function; therefore the treatment will be limited to entropy *changes* ($d\phi$, $\delta\phi$, $\Delta\phi$), rather than to consider the *absolute* amount.

(**b**) To a student unable to distinguish between heat and cold and not familiar with the phenomena accompanying temperature changes, it would be very difficult to convey a conception of what a temperature change really is. Probably the best definition would be the mathematical one $\delta T = \frac{\delta Q}{C}$, which would be unsatisfactory and troublesome to the student until, by experience, he became familiar with the phenomena accompanying changes in temperature. The same difficulty occurs in attempting to define any unfamiliar physical quantity or property, and applies equally well to entropy. Hence, the best

that can be done at present is to give a mathematical definition of entropy and rely on the experience and familiarity, which will come from the solution and discussion of problems involving its use, to give a more or less concrete conception of the physical meaning and properties.

(c) **The Mathematical Expression for an Infinitesimal Change of Entropy** is

$$d\phi = \frac{dS + dI + AP\,dV}{T}, \quad . \quad . \quad . \quad . \quad . \quad (53)$$

in which the numerator indicates the summation of the infinitesimal changes indicated,

T = absolute temperature of material during these infinitesimal changes, and

A = 1/778, introduced to keep all terms in numerator in same units.

A finite change of entropy will then be

$$\Delta\phi = \int_1^2 \frac{dS + dI + AP\,dV}{T} \quad . \quad . \quad . \quad . \quad . \quad (54)$$

It is often convenient to first evaluate per pound and then for the weight concerned. If $\Delta\phi_1$ is for unit weight,

$$\Delta\phi_W = W\,\Delta\phi_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

(d) It will be observed during the further development of thermodynamic phenomena that all those processes which occur " naturally," i.e., spontaneously, are accompanied by an increase of entropy. Any process which results in a decrease of entropy must be forced in some way, and is in that sense " unnatural." Hence it may be said that the entropy of every substance tends to increase.

A somewhat analogous, though not a parallel, case may be cited from mechanics. It is well known that the potential energy of mechanical systems always tends to decrease, for there is a tendency for the centers of mass of all terrestial bodies to approach the earth's center as closely as conditions will permit. Given a mechanical system, in which processes resulting in change of the position of the center of gravity can take place, that change will occur which will make the potential energy of the system least, unless external forces impose a different behavior.

38. Entropy Changes for Reversible Processes with Ideal Gases. (a) It was mentioned (in Section 34c) that the ideal expansions considered in Chap. V may be made reversible processes; and a further consideration will show that in every such case the external work, dE, can be represented as $AP\,dV$. Further study will show that for all reversible processes

$$AP\,dV = dE \text{ and } A\int_1^2 P\,dV = \Delta E.$$

Then, since $dI = 0$ for an ideal gas, the numerator of Eq. (53) becomes $dS + dE = dQ$. And the infinitesimal entropy change experienced by an ideal gas, during a reversible process, is then

$$d\phi = \frac{dS + dE}{T} = \frac{dQ}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (56)$$

The finite change is

$$\Delta\phi = \int_1^2 \frac{dQ}{T}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

or, if dQ be assumed to refer to unit weight, see Eq. (55),

$$\Delta\phi_W = W\int_1^2 \frac{dQ}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (58)$$

(b) It must be noted that the last three equations are proved only for *reversible* processes, and for the present they will be considered as applicable only to ideal gas. They may be used for finding the entropy change experienced by a given weight of ideal gas which, while expanding behind a piston, is undergoing one of the reversible processes, such as those described in Chapter V; they are not applicable for determining the entropy change when a given weight of gas experiences an irreversible process, such as the free expansion of Joule's experiment, Section 35 (b). In such cases the entropy change must be determined in other ways which will be presented later.

39. Sign of Entropy Changes during Reversible Processes. (a) The integration of Eq. (57) between the limits 1 and 2 for any assumed process will result in a difference of two quantities; thus the sign of the right-hand member will depend upon which of these quantities is the larger. The sign of this number indicates whether the process in question will increase or decrease the entropy of the material.

(b) A reversible increase of heat energy would give a positive value for the right-hand member of the equation, — a positive value of $\Delta\phi$, — and this corresponds to an increase of entropy.

(c) A reversible rejection of heat results in a negative value of $\Delta\phi$, indicating a decrease of entropy.

(d) Eq. (57) would show no change of entropy for any reversible process involving no change of associated heat, but this could only be true of an adiabatic process (in which $dQ = 0$, see Section 30 (a)). An ideal gas, therefore, experiences no change of entropy during a reversible adiabatic process.

(e) Although it is the entropy *change* which is really considered in thermodynamic calculations in which entropy is concerned, yet engineers are accustomed to speak of the "*total entropy*" of the substance for the particular conditions of temperature, pressure, and volume pertaining. They do this because they have by common consent agreed that the entropy of materials shall be measured above a certain arbitrarily chosen datum, which is taken as zero for convenience. Thus the term "total entropy" (ϕ) refers to the total entropy *change* experienced by the material in passing reversibly from the arbitrarily chosen datum to the conditions in question. As the entropy difference ($\phi_2 - \phi_1$) is dealt with, any datum whatever may be selected provided the same one is used for both of the entropy quantities, ϕ_2 and ϕ_1.

40. Entropy Changes during Reversible Isobarics of Gases. In Eq. (56) the numerator of the right-hand member can be replaced by the product of specific heat into an infinitesimal temperature change, thus *for unit weight*

$$d\phi = \frac{dQ}{T} = \frac{C\,dT}{T}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (59)$$

the symbol C representing the proper specific heat for the particular change under consideration.

For a change at constant pressure C becomes C_p and the differential equation for entropy change, *per unit weight*, is

$$d\phi = \frac{C_p\,dT}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

Assuming C_p a constant as before, the total change of entropy is, *per unit weight*,

$$\int_1^2 d\phi = C_p \int_1^2 \frac{dT}{T}. \quad . \quad . \quad . \quad . \quad . \quad (61a)$$

Thus $$\Delta\phi = (\phi_2 - \phi_1) = C_p (\log_e T_2 - \log_e T_1). \quad (61b)$$

$$= C_p \log_e{}^* \frac{T_2}{T_1}. \quad (61c)$$

Eq. (61b) or (61c) will indicate by the algebraic sign of its right-hand member whether a positive or negative entropy change is under consideration. Increase of associated heat will make T_2 greater numerically than T_1 and the right-hand member of the equations will then have a positive sign, which indicates an increase of entropy. Reduction of associated heat will make T_2 less than T_1 and the right-hand member of the equations will have a negative sign.

The equations can then be trusted to give not only the numerical value of the entropy change, but also to signify whether it increases or decreases the total entropy of the material under consideration.

41. Entropy Changes during Reversible Isovolumics of Gases. In this case the specific heat C in Eq. (59) becomes C_v and the resulting differential equation is

$$d\phi = \frac{C_v\, dT}{T}. \quad (62)$$

The total change of entropy is, therefore, *per unit weight*,

$$\int_1^2 d\phi = C_v \int_1^2 \frac{dT}{T},$$

or $$\Delta\phi = (\phi_2 - \phi_1) = C_v (\log_e T_2 - \log_e T_1). \quad (63)$$

$$= C_v \log_e \frac{T_2}{T_1}. \quad (63a)$$

As before, the algebraic sign of the right-hand member of this equation will indicate whether an increase or decrease of entropy is under consideration.

42. Entropy Changes during Reversible Isothermals of Gases. During an isothermal change the temperature is constant by definition; that is, T in Eq. (56) is the same for each of the differential heat changes dQ. Then

$$\int_1^2 d\phi = \int_1^2 \frac{dQ}{T}$$

* It is usually more convenient to use $\log_{10}$ instead of $\log_e$. As $\log_e = 2.302 \log_{10}$, Eq. (61c) may be written $\Delta\phi = C_p \times 2.302 \log_{10} (T_2/T_1)$. The other logarithmic equations which are to follow may be similarly transformed.

becomes

$$\Delta\phi = (\phi_2 - \phi_1) = \frac{(Q_2 - Q_1)}{T} \quad . \quad . \quad . \quad . \quad . \quad (64)$$

Thus, for isothermal changes, since $(Q_2 - Q_1) = \Delta Q$,

$$\Delta\phi = \frac{\Delta Q}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

The entropy change will obviously have the same sign as ΔQ, indicating increase of entropy with increase of associated heat and decrease of entropy with decrease of associated heat.

43. Entropy Changes during Reversible Adiabatics of Gases. An adiabatic change being one which occurs under conditions of heat insulation, that is, one during which heat energy is neither given to nor abstracted from the substance, it follows that $dQ = 0$, and therefore

$$d\phi = \frac{dQ}{T} = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

Thus during a reversible adiabatic change there is no entropy change, just as during an isothermal process there is no temperature change. *Reversible adiabatics* are therefore often called *Isentropics*, and these two terms may be used interchangeably.

44. Irreversible Adiabatic Processes of Ideal Gas, and the Corresponding Entropy Changes. (a) Besides the *reversible* adiabatic expansion already discussed, there are an unlimited number of adiabatic processes which are irreversible. These are thermodynamic processes which ideal gas undergoes when confined in vessels which are nonconducting as regards heat, that is, in those which neither permit the gas to receive nor to surrender any heat through the surrounding walls. Of the processes which are strictly adiabatic, those which are isentropic are the only ones that are reversible.

(b) As an example of an irreversible adiabatic change of ideal gas, assume the process similar to the free expansion of Joule's experiment, discussed in Section 35 (b). During such a process, the entropy change experienced by unit weight of gas cannot be found by Eqs. (56) and (57), as they apply only to reversible processes. An attempt to use them would give the zero entropy

change that was obtained in Section 43, which is very far from being correct, as the next paragraph will show.

(c) Recourse must then be had to the original definition, Eq. (53), which may be rewritten as

$$d\phi = \frac{dS + dI}{T} + A\frac{PdV}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (a)$$

In the process under discussion there can be no change in the sensible heat because the temperature of the gas is the same after as before the change, and dI is of course zero for ideal gas. Hence $dS + dI = 0$ and Eq. (a) becomes

$$d\phi = A\frac{PdV}{T}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (b)$$

The PVT changes in a *unit* weight of ideal gas are represented by the expression

$$\frac{P\mathbf{V}}{T} = R.$$

Thus $$P = RT/\mathbf{V}$$

and $$Pd\mathbf{V} = RT\frac{d\mathbf{V}}{\mathbf{V}}.$$

This value of $Pd\mathbf{V}$ may now be substituted in Eq. (b), which then becomes

$$d\phi = A\frac{RT}{T}\frac{d\mathbf{V}}{\mathbf{V}} = A\frac{R\,d\mathbf{V}}{\mathbf{V}}.$$

Integrating this between the limits 1 and 2 gives the true entropy change, per pound of material,

$$\Delta\phi = A\int_1^2 R\frac{d\mathbf{V}}{\mathbf{V}} = AR(\log_e \mathbf{V}_2 - \log_e \mathbf{V}_1). \quad . \quad . \quad (c)$$

As the volume $\mathbf{V}_2$ occupied when the gas fills the two vessels is greater than the volume $\mathbf{V}_1$ which it had when confined in one of them, the process, as shown by Eq. (c), must result in an increase of entropy despite the fact that the conditions are adiabatic. This is quite different from the zero value obtained by applying the equation for reversible changes in which $APdV = dE$. This emphasizes the statement already given that only *reversible* adiabatics are isentropic processes.

(d) The free expansion of a gas may be called a "natural" process. It was seen to be accompanied by an *increase* in

entropy of the materials concerned. A similar increase also occurs with all other natural processes, such as the flow of heat from a higher temperature body to one at lower temperature. Thus the entropy of all substances always tends to increase. These facts will become more apparent as the subject is developed.

45. Entropy Changes Independent of Path. (a) The integration of Eq. (54) results in a difference in two quantities, the values of which depend merely on the conditions of the substance before and after the change. Evidently, then, the entropy change is in no way dependent on the method of changing from the one set of conditions to the other. Thus the entropy change experienced by a material in passing from some definite set of conditions 1 to another definite set 2 will always be the same, no matter what path is pursued on the graphical representation of the process.

(b) This fact is often of great importance, as the entropy change experienced by a substance when undergoing any very complicated set of changes can be determined by finding the entropy change accompanying any simple change, or group of changes, which will carry the body from the same initial to the same final conditions. It is, however, very essential to make sure that the final conditions *are* the same in both cases, as mistakes are easily made in just this point.

46. Temperature-Entropy Diagrams. (a) Just as pressure-volume diagrams are useful as a means of graphically representing certain changes, so diagrams with absolute temperature plotted vertically and entropy change plotted horizontally are capable of visualizing some very important transformations. They are known as **Tϕ-diagrams**.

There is a peculiarity about the plotting of diagrams with temperature and entropy coördinates, to which attention should be called. In the PV-diagram the intersection of the coördinate axes represents zero pressure and zero volume, and this is possible because both absolute pressure and absolute volume can be determined. In the Tϕ-diagrams, however, although absolute temperature can be determined inferentially, as previously shown on p. 30, the *absolute quantity of entropy is indeterminate* like the absolute quantity of associated heat. As already shown, the equations give *change of entropy*, $\Delta\phi$, and ***not***

absolute quantity of entropy, ϕ. It is this $\Delta\phi$ which is used in plotting. The abscissas thus represent entropies of a substance *above some conveniently chosen datum*, such as that at 32° F.

(b) In Fig. 15, the point A represents the temperature-entropy conditions of a substance as $T_1\phi_1$. This means that at temperature T_1 the entropy of the substance is ϕ_1, above what-

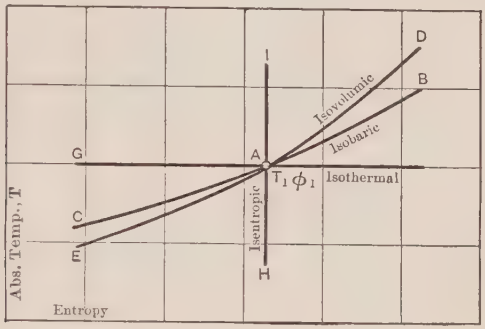


Fig. 15. — Tϕ-Curves for Gases.

ever value has been decided on as zero of entropy, — just as in the PV-diagram, Fig. 6, V_1 represents the volume of the substance above absolute zero of volume when its pressure is P_1.

Isobaric Changes.

(c) The line AB, Fig. 15, represents the temperature-entropy changes of a gas expanding at constant pressure, or it is the graph of Eq. (61), and is obtained by substituting various increasing values for T_2. Similarly, the line AC is the graph of a constant-pressure compression.

Isovolumic Changes.

(d) The line AD represents a rise of pressure at constant volume and is obtained by means of Eq. (63); while the line AE is the same curve continued backward, and represents a constant-volume pressure drop.

Isothermal Changes.

(e) During an isothermal change T is constant but entropy becomes greater as associated heat increases, which occurs as volume grows larger. The graph of an *isothermal expansion*

from $T_1\phi_1$ must then be a *horizontal line* to the right of A; and similarly an isothermal compression must be shown by the horizontal line to the left.

Adiabatic Changes.

(f) The entropy change is zero during *a reversible adiabatic change*, therefore $\Delta\phi$ equals zero, and such a change must be shown by a *vertical line* on the Tϕ-diagram. Further, since the temperature of a gas decreases during adiabatic expansion, as previously shown, the line $A\,H$ must represent such an expansion from $T_1\phi_1$ conditions, and the line $A\,I$ a similar compression from the same point.

Area equivalent to ΔQ.

(g) From Eq. (56) $dQ = T\,d\phi$, for reversible processes, and hence for such a process,

$$\int_1^2 dQ = \int_1^2 T\,d\phi,$$

and

$$\Delta Q = Q_2 - Q_1 = \int_1^2 T\,d\phi. \quad . \quad . \quad (67)$$

The last term of this equation is, however, the mathematical expression for the area under a curve drawn to $T\phi$-coördinates. It therefore follows that *area on the $T\phi$-diagram represents heat change* during reversible processes, and inspection of the graphs already given will show that area under a line traced from left to right represents heat given *to* a substance, while area under a line from right to left represents heat abstracted *from* a substance.

One of the great conveniences resulting from the use of the Tϕ-diagram in engineering may now be seen. The PV-diagram shows by the area beneath the expansion line the total *external work* associated with a process; while, for reversible processes at least, the Tϕ-diagram shows by the area beneath the corresponding line the change of total *associated heat* occurring during the same processes. The engineer is thus enabled to quickly solve many problems by simple inspection of these two diagrams, and can avoid the necessity of making long mathematical calculations with the ever-present possibility of error.

It may be objected that it involves more work and time to construct the necessary diagrams than it would to make the

calculations direct. It will, however, be discovered in a later chapter that certain standard diagrams can be constructed for the solution of by far the larger class of problems in which the conception of entropy change is particularly helpful. These diagrams, once constructed, can be used indefinitely without further calculation.

CHAPTER VIII.

GAS CYCLES.

47. Definition of a Cycle. (a) As already stated, man requires far more energy than his body can supply, and this energy is obtained from Nature's stores. Energy as used by the engineer is always associated with some substance, body, or "system": kinetic energy with moving masses; potential mechanical energy with masses by virtue of position; heat, sensible or latent, with solids, liquids, or gases.

(b) When energy is used for the doing of work the material with which it is associated is called the **working substance.** Thus in a hydraulic power plant, water is the working substance; gas is the working substance in a gas engine; and water is the working substance in a steam engine.

(c) If a given quantity of a working substance, with its associated energy, be used so as to obtain *all the work possible* under given circumstances, the same amount of work cannot be again obtained under the *same circumstances* unless the substance is first returned to its *initial condition.* Thus a pound of water falling a given distance will develop a certain amount of work, and that work will be the greatest obtainable under the circumstances when the water falls to the lowest possible point. To again develop the same amount of work with the same pound of water, it must first be raised to the height from which it originally fell. Or similarly, if one pound of gas does work by expanding adiabatically from a temperature T_1 to a temperature T_2, which is the lowest possible under the conditions, it cannot again do the same amount of work in the same way until its temperature is again raised to the initial value T_1.

(d) In order to deliver work *continuously* as is generally required, there are obviously only two possible methods of operation: either (1) *the working substance must be periodically returned to initial conditions,* or (2) *new quantities of working*

substance must be supplied at regular intervals. The latter is the simpler and is often used, but it is not Nature's method. If man uses falling water to develop power and allows the water to run to waste at the lower level, Nature immediately begins to lift it by evaporation, so that sooner or later it will be again available. If man burns carbon in air, getting hot CO_2 and N_2, and then, after obtaining work by lowering the temperature, discharges the gas as useless, Nature through the agency of plant growth decomposes the cold CO_2 into C and O_2 so that they can again be combined to evolve the same amount of heat energy as before.

Thus without man's agency all working substances periodically return to the same starting conditions, that is, pass through *cycles.*

A **cycle** *is any series of operations which periodically brings the working substance back to initial conditions.*

It is customary to speak of **Open** and **Closed Cycles**, but there are really no open cycles in engineering. If the engineer carries a working substance through any series of changes which does not return it to initial conditions, Nature kindly closes the cycle for him.

(e) One difficulty here confronts the beginner: Experience shows that it requires at least as much energy to pump water between levels as can be obtained from it in flowing down again; this being true, how is man to obtain available work from a substance if equal work has to be returned to raise the material to starting conditions? There are two solutions which amount to the same thing in the end:

1. Allow Nature to do the pumping, as in the case of the waterfall; or

2. Imitate Nature in finding some way of pumping that does not require the return of the identical energy which has been obtained from the cycle.

When a heat engine is used *heat energy is available* but *mechanical energy is sought.* Most of the methods in use for returning the working substance to initial conditions in such cases depend upon the use of a small amount of the generated mechanical energy and a large amount of the available heat energy for this purpose; or they employ some group of processes which are the substantial equivalent of this.

48. Diagram of a Cycle. (a) Cycles are conveniently represented diagrammatically, as has already been done for pressure-volume changes or temperature-entropy changes. The coördinates used are generally either PV, or $T\phi$.

Assume for instance that the point A, in Fig. 16, represents the pressure and volume conditions P_1V_1, at temperature T_1, of *one pound* of gas used as a working substance in a cylinder fitted with a frictionless piston, as shown in the figure. If the gas expands to conditions P_2V_2 at B, according to such a law that AB is the graph of pressure-volume changes, the area $ABEF$ must measure the external work done upon the piston while it moves from position a to position b. If the gas then expands further according to some other law BC so as to arrive at the point C with conditions P_3V_3, the additional external work done upon the piston while moving from b to c must be represented by the area $BCGE$.

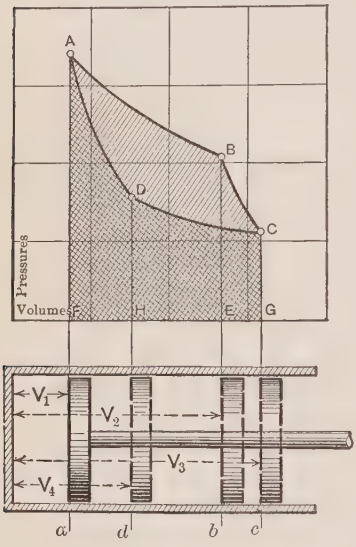


Fig. 16. — PV-Diagram of Cycle.

By compression the working substance may then be brought to some conditions P_4V_4 according to the law represented by the graph CD while the piston moves from c to d, but, to do this, work represented by the area $CDHG$ must be done by the piston upon the gas. From point D compressing in the proper way will bring the working substance to starting conditions at A, with an expenditure of work shown by the area $DAFH$. The return of the gas to starting conditions at A completes a cycle; the pressure, volume, and temperature of the gas are again $P_1V_1T_1$ and the piston is back to position a. There is then no reason why the same process may not be repeated again and again indefinitely. Observe, however, that the total external work done *by* the gas is

$$\textit{Positive Work} = ABEF + BCGE = ABCGF \text{ ft.-lbs.},$$

while the total work done *upon* the gas is

$$\textit{Negative Work} = CDHG + DAFH = CDAFG \text{ ft.-lbs.}$$

leaving

$$\begin{aligned}\textit{Net or Available Work} &= ABCGF - CDAFG \\ &= ABCDA \text{ ft.-lbs.} \\ &= \textit{Area inclosed by lines of cycle.}\end{aligned}$$

(b) Four successive processes as represented by the four lines in Fig. 16 are not necessary for a working cycle. Any number of processes between an infinite number and two may inclose an area and therefore could represent a cycle delivering work. Four lines are, however, employed in most of the cycles used in ordinary heat engines.

49. The Carnot Cycle for Gases. (a) This cycle, named from Sadi Carnot, the man who first investigated it, represents the **best that can possibly be done in the conversion of heat energy into mechanical energy.*** It cannot be used in any actual engine and is therefore only of *theoretical interest as a criterion of the maximum result obtainable.*

(b) For performing such a cycle with gas it is necessary to have

1. The gaseous working substance;
2. Certain apparatus, to be specified below.

The working substance may be any gas far enough removed from its point of liquefaction to sensibly obey the laws already developed for perfect gases.

The necessary apparatus is shown in Fig. 17 and may be described as follows:

U is a body at high temperature T_1 and so arranged that this temperature remains constant despite withdrawal or addition of heat energy. An ordinary furnace with a controllable fuel and

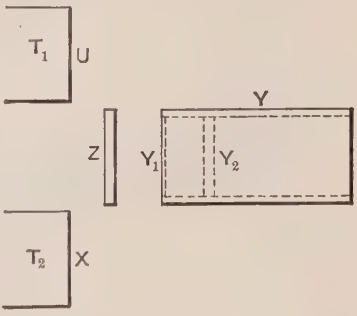


Fig. 17. — Machinery of Carnot Engine.

* This statement must not be interpreted to mean that no other cycle can do as well; it means only that no other cycle can do better. It will be shown later that there are a number of cycles equally efficient as energy transformers.

air supply approximates these conditions. The body U will hereafter be called the **Hot Body.**

X is a body at temperature T_2, lower than T_1, and this temperature T_2 remains constant despite addition or removal of heat energy. A vessel jacketed with flowing water at temperature T_2, or arranged like the condenser shown in Fig. 3, would approximate these conditions. The body X will hereafter be called the **Cold Body.**

Y is a cylinder, Z is a removable plate which may be used to cover the end of the cylinder, and Y_2 is a frictionless piston. These parts are made of material that will neither absorb nor conduct heat. Y_1 is a cylinder head made of material that offers no resistance to flow of heat.

Operation of Carnot Engine.

(c) Imagine first that *one pound of gas* is inclosed in the cylinder Y at conditions $P_a\mathbf{V}_a$ and T_a, as shown at a in Fig. 18, T_a being equal to T_1, the temperature of the hot body. (1) Remove cover Z, apply the hot body to the conducting head Y_1, and allow the gas to *expand isothermally* to some lower pressure P_b at volume $\mathbf{V}_b$ as shown at b in the figure. The necessary heat supply must have come from the hot body and may be called ΔQ_1.

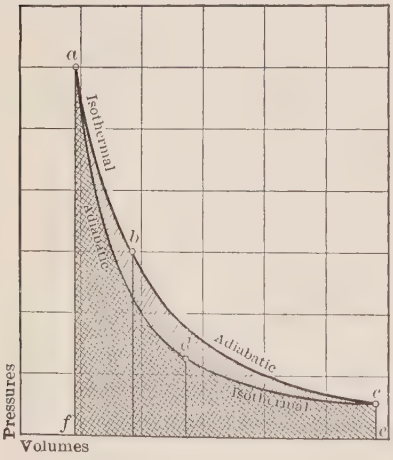


Fig. 18. — PV-Diagram of Carnot Cycle.

(2) Next remove the hot body, apply the nonconducting cover Z, and allow the gas to *expand adiabatically* until its temperature has dropped to T_c, equal to T_2, the temperature of the cold body. The gas will then have conditions $P_c\mathbf{V}_c$.

(3) Again remove the cover Z, apply the cold body X, and drive the piston back, *compressing* the gas *isothermally* to some higher pressure P_d at volume $\mathbf{V}_d$. (The value of P_d will be con-

sidered in the next paragraph.) The heat generated must be absorbed by the cold body and may be called ΔQ_2.

(4) For the fourth and completing operation, remove the cold body, replace the nonconducting head Z, and drive the piston back, *compressing* the gas *adiabatically* until its temperature has again risen to that of the hot body, which was the starting temperature of the cycle. To close the cycle, the pressure and volume must return to $P_a V_a$ when T_1 is reached. This can only be attained if the isothermal compression is stopped at such a point d that the subsequent adiabatic compression will return the gas to the starting conditions.

Work Developed by Carnot Engine.

(d) The area crosshatched upward from left to right in Fig. 18 represents work done *by* the gas, while that crosshatched downward from left to right represents work done *upon* the gas. The foot-pounds of **net work** resulting from one cycle is shown by the inclosed area *abcd*. If this cycle is carried through n times per minute, the total net work done by the gas will be n times the area *abcd*. The mathematical expression for net work done per cycle can be obtained by using the formulas already developed for isothermal and adiabatic changes. The results are tabulated below.

Before consulting this table, however, note that this cycle consists of two isothermals joined by two adiabatics. The T_1 isothermal is an expansion with ratio $\frac{V_b}{V_a} = r$, and the T_2 isothermal is a compression with ratio $\frac{V_c}{V_d} = r'$. These two ratios must be equal because by Eq. (51)

$$\frac{T_1}{T_2} = \frac{T_b}{T_c} = \left(\frac{V_c}{V_b}\right)^{\gamma-1}$$

and

$$\frac{T_1}{T_2} = \frac{T_a}{T_d} = \left(\frac{V_d}{V_a}\right)^{\gamma-1}$$

giving

$$\frac{V_c}{V_b} = \frac{V_d}{V_a}, \quad \text{or} \quad \frac{V_b}{V_a} = \frac{V_c}{V_d},$$

so that

$$r = r'.$$

By means of the last equation the tabulated results give simple expressions for net work as indicated below the table.

Line.	Kind.	Heat Received (Ft.-lbs./lb. gas).	Work Done (Ft.-lbs./lb. gas).
ab	Isothermal Expansion	$+ RT_1 \log_e r$	$+ RT_1 \log_e r$
bc	Adiabatic Expansion	0	$+ R\frac{T_1 - T_2}{\gamma - 1}$
cd	Isothermal Compression	$- RT_2 \log_e (r' = r)$	$- RT_2 \log_e (r' = r)$
da	Adiabatic Compression	0	$- R\frac{T_1 - T_2}{\gamma - 1}$

$$\text{Net Work} = RT_1 \log_e r - RT_2 \log_e r'$$
$$= (T_1 - T_2) R \log_e r \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (68)$$

Efficiency of Carnot Engine.

(e) **Efficiency is defined** *as the ratio of useful result to expenditure or effort made to obtain that result.* That is

$$\text{Efficiency} = Ef. = \frac{Result}{Effort}.$$

The result obtained from the operation of this Carnot engine is the *net work done by the gas* and the expenditure made is the *heat supplied by the hot body.* Then

$$\text{Ef. Carnot Cycle} = \frac{Foot\text{-}pounds\ represented\ by\ abcd}{778\ \Delta Q_1}. \quad (69a)$$
$$= \frac{B.t.u.\ represented\ by\ area\ abcd}{\Delta Q_1}. \quad (69b)$$

The heat supplied per unit weight of gas is $\Delta Q_1 = RT_1 \log_e r$ foot-pounds and the net work is given by Eq. (68). Hence:

$$Ef. = \frac{Net\ Work}{Heat\ Supplied} = \frac{(T_1 - T_2) R \log_e r}{T_1 R \log_e r} = \frac{T_1 - T_2}{T_1}. \quad (69c)$$

Objection is often made to the expressions of efficiency just developed because it seems as though the engine ought to be credited with the heat given to the cold body. The fallacy of this appears when it is understood that the heat given to the cold body leaves the engine at a *low temperature*, T_2, whereas to operate the engine heat must be available *at a high temperature* T_1. The heat rejected to the cold body *could not*, therefore, *be*

directly used again in the engine,* and hence should not appear in the expression for efficiency.

(f) To make this heat available again for use in the same engine, it would have to be raised to the high temperature T_1 and be returned to the engine by way of the hot body at that temperature, but *experience shows that heat will not flow of its own accord from any body to one at a higher temperature.* From the discussion which follows, it will be seen that *at least as much mechanical energy would be consumed in causing such a flow as could be obtained by using the elevated heat in a heat engine.* It will be discovered that this is all in accord with the Second Law of Thermodynamics.

The case is analogous to that in which water leaving a water wheel is pumped again to the original height in the attempt to utilize the energy still possessed by the water when leaving the wheel. Obviously, in this case the energy leaving the wheel with the effluent water is of no further use to that wheel, and exactly the same thing is true of the heat energy leaving the engine.

(g) Fig. 19 is intended to show the energy changes of the Carnot cycle graphically. If vertical distances between heat reservoirs T_1 and T_2 in the figure represent temperature, and widths of streams represent quantities of energy, the sense of the foregoing discussion becomes graphically evident.

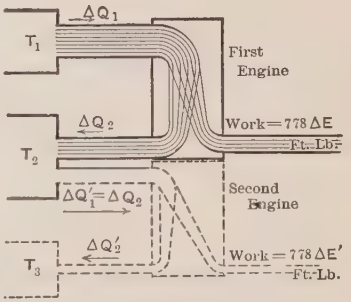


Fig. 19. — Energy Flow in Carnot Engine.

The dotted part of the figure shows how a part of the effluent energy might be used if another cold body with temperature T_3, lower than T_2, could be obtained.† The ultimate limit to this

* In a real case the hot body would derive the heat to maintain its temperature from some form of fuel, and the cost of that fuel would be the expenditure made to obtain the work delivered by the engine.

† The possibility of the existence of cold body T_3 immediately suggests the use of only one engine operating between temperatures T_1 and T_3. There is no theoretical objection to this, but sometimes when analogous schemes are tried with real engines a number of practical considerations dictate the use of several engines in series as above, rather than one engine working through the entire temperature drop. The reasons will be considered later.

arrangement would be an engine having a cold body with temperature at absolute zero.

It is of interest to note that *in this limiting case the Second Law of Thermodynamics would no longer be true* because the last engine of the series would reject no heat, having reduced the temperature of its working substance to absolute zero. All the heat entering the group of engines could then be completely and continuously converted into mechanical energy. *It is obviously an impossible proposition*, arising in this case from the impossible ideal gas, the assumptions made as to the properties of that material, and the absurd assumption that any body can be maintained indefinitely at absolute zero of temperature without the expenditure of work in a continuous process of refrigeration.

From Fig. 19

$$\Delta Q_2 + \Delta E = \Delta Q_1; \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

hence the efficiency might be written,

$$Ef. = \frac{\Delta E}{\Delta Q_1} = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1}, \quad . \quad . \quad . \quad . \quad (71)$$

and this will be found to express the efficiency of any heat-engine cycle. From Eqs. (69c) and (71) it is evident that in the case of the Carnot engine with gaseous working substance

$$\frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1} = \frac{T_1 - T_2}{T_1}. \quad . \quad . \quad . \quad . \quad . \quad (72)$$

Reversibility of Carnot Engine.

(h) **Each part of the process carried on in a Carnot engine is thermodynamically reversible.** In fact the cycle is made up of the two processes which were cited in Section 34 (a) and (b) as typical examples of reversible processes. **The entire cycle must therefore be reversible**; that is, it must be possible to operate the engine starting at the point *a* in Fig. 18, and following the cycle backwards in the direction *adcb*.

There is no reason why the gas cannot (1) *expand adiabatically* from *a* to *d* and then (2) *isothermally*, at temperature T_2, from *d* to *c*. During the latter process it would *absorb heat* ΔQ_2 from the cold body. If the necessary mechanical energy is available the gas can be (3) *compressed adiabatically* from *c* to *b*, and then (4) *isothermally* in contact with the hot body to the starting point. During the isothermal compression the gas *must give to*

the hot body the amount of heat ΔQ_1 exactly *equal to that previously removed during the direct operation.*

In the diagram (Fig. 18) the work done by the gas during the two expansions must be represented by the area *adcef*, and that done on the gas during the two compressions must be shown by the area *ecbaf*. The net result must then be the *absorption of external work equal to that given out* in the direct cycle and represented by the area *adcb*. Tabulation of the results of operation, first direct and then reversed, gives

Direct Operation.		Reversed Operation.
Heat absorbed from hot body =	ΔQ_1	= Heat delivered to hot body.
Heat discharged to cold body =	ΔQ_2	= Heat absorbed from cold body.
Mechanical energy delivered =	$(\Delta Q_1 - \Delta Q_2)$	= Mechanical energy absorbed.

Thus the Carnot engine reversed can remove heat at low temperature from the cold body and, *having absorbed a certain quantity of available mechanical energy*, can deliver the sum of these two energies to the hot body as heat at high temperature. It is therefore a heat pump.

Carnot Engine as a Source of Perpetual Motion of Third Type.

(i) Imagine now two Carnot engines exactly alike, one working as an engine, and the other, with operation reversed, working as a "heat pump." The engine will remove heat from the hot body, deliver a part of it as mechanical energy, and discharge the remainder to the cold body. The pump will absorb from the cold body the same quantity of heat that this latter received from the engine; it will require for its operation the same quantity of mechanical energy that was delivered by the engine; and it will discharge to the hot body the sum of the two energies, — that is, an amount of heat equal to that which the engine removed from the hot body. If the two pieces of apparatus can be connected so that the engine drives the pump, a device results which, theoretically devoid of friction and radiation losses, can go on moving forever, though delivering no useful work. This is what was called in Section 9 perpetual motion of the third type, which, though conceivable, cannot be materialized.

50. All reversible engines have the same efficiency as the Carnot engine when working between the same temperature limits.

There are many possible types of reversible and irreversible ideal engines. It will now be proved that, when working between the same temperature limits, — i.e., receiving heat from a hot body at the same temperature as that supplying the Carnot engine and rejecting heat to a cold body at the same temperature as that used with the Carnot engine, — (1) *no engine whatever can have higher efficiency than the Carnot engine and* (2) *the efficiency of all reversible engines equals the efficiency of the Carnot engine.*

To prove (1): Assume that any engine, A, is more efficient than the Carnot engine, C. Obviously A could deliver more mechanical energy than could C, although receiving the same amount of heat; and the heat rejected by A to the cold body would evidently be less than that delivered by C by an amount equal to the difference between the quantities of mechanical energy delivered by A and C.*

Let A, operating as an engine, drive C reversed, that is, as a heat pump. This is shown diagrammatically in Fig. 20, in

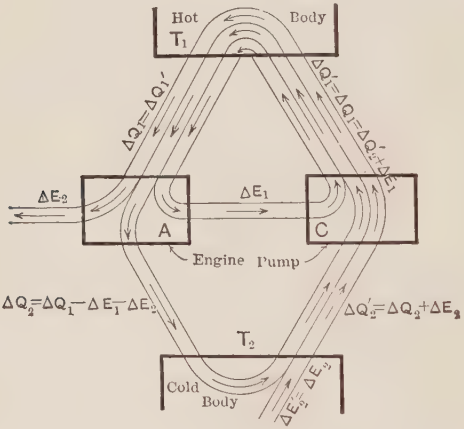


Fig. 20. — Heat Flow Diagram to show that no engine can have a greater efficiency than the Carnot.

which the width of stream is supposed to be a measure of the energy flow. From the combination there would result an excess of mechanical energy ΔE_2 which could be used outside

* Because heat received = heat discharged + mechanical energy delivered. With the left member of the equation constant, neither term of the right member can vary except at the expense of the other.

the system. This excess mechanical energy would be exactly equal to the *only heat supplied the system*, that is, to $\Delta E_2'$ given by the cold body. Therefore the combination could *continuously* convert into mechanical energy *all* the heat supplied it; but this would be Perpetual Motion of the Second Type, and is contrary to human experience as expressed in the Second Law of Thermodynamics. Since the assumption that A is more efficient than C results in this impossibility, it follows that that assumption must be incorrect, and that no heat engine, reversible or irreversible, can have an efficiency greater than that of the Carnot engine. Hence statement (1) is correct.

To prove (2), that if the engine A is reversible it must have the same efficiency as the reversible Carnot engine C working between the same temperature limits, imagine it to have an efficiency less than that of C.* Being reversible, it can be used as a heat pump driven by C. Then, if the pump can be less efficient than the engine, perpetual motion of the second type again appears. Hence, neither engine can be more efficient than the other, so the efficiencies of all reversible engines working between the same temperature limits must be the same, which proves (2).

51. Comparision of Carnot Engine and Real Engine. The Carnot engine as described above is evidently only an ideal mechanism, for the material assumed for the parts does not exist and a perfect apparatus could not be constructed. It is possible, however, to approach such ideals and they may therefore be regarded as limiting cases for actual constructions. Comparing the real engines with the Carnot as a standard gives a measure of the perfection of attainment.

In any actual engine, the piston, which itself meets with frictional resistance, is connected to a friction-burdened mechanism. In the real engine, provision must also be made for storing part of the energy delivered during the expansion, to be returned for the compression of the working substance, and this storage and return always involves waste. In the reciprocating engine, for instance, this energy-storing device is usually a flywheel, and some of the energy stored is lost in friction and windage.

* It has already been proved that its efficiency cannot be greater than that of C.

Obviously there must be the following losses in any real engine:

1. Some of the heat received from the hot body must be lost as heat through conduction and radiation by the material of the engine.

2. Some of the mechanical energy delivered to the piston must be lost by friction in the mechanism of the engine.

3. Some of the energy stored for compressing the working substance must be lost by friction during its storage and its return.

Recalling the meanings of the three types of perpetual motion (page 7), it is evident —

(*a*) *That no ideal engine can give perpetual motion of the first or second type;*

(*b*) *That any ideal reversible engine combined with another of similar character can give perpetual motion of the third type only;* but

(*c*) *That no real engine can give perpetual motion of any of the three types.*

52. Tϕ-Diagram of Carnot Cycle. (**a**) The Carnot cycle, being made up of *two reversible isothermals* and *two reversible adiabatics*, must be represented by a *rectangle* when drawn to Tϕ coördinates. Such a diagram is given in Fig. 21, in which the horizontal lines *ab* and *cd* represent the reversible isothermal reception and rejection of heat and the vertical lines *bc* and *da* show the reversible adiabatic changes. The corresponding corners of the cycle are for convenience lettered the same as in Fig. 18.

(**b**) Since for reversible changes with ideal gases,

$$\Delta Q = \int_1^2 T\, d\phi$$

the area *abef* under the isothermal expansion *ab* represents heat ΔQ_1 received from the hot body, and the area *cefd* is similarly proportional to heat ΔQ_2 rejected to the cold body during the isothermal compression *cd*. The difference *abcd* is the *area of the cycle and represents heat converted into work.* Then

$$\Delta Q_1 = T_1(\phi_b - \phi_a) \quad \text{and} \quad \Delta Q_2 = T_2(\phi_c - \phi_d) = T_2(\phi_b - \phi_a)$$

and

$$Ef. = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1} = \frac{T_1(\phi_b - \phi_a) - T_2(\phi_b - \phi_a)}{T_1(\phi_b - \phi_a)}$$

$$= \frac{T_1 - T_2}{T_1}, \text{ as before.}$$

53. Criterion of Maximum Efficiency. That an ideal engine may have the maximum possible efficiency, $\frac{T_1 - T_2}{T_1}$, when receiving heat from a body at temperature T_1 and rejecting heat to a body at temperature T_2, it is necessary that —

(1) *All heat received from the hot body be taken when the working substance has the same temperature as that body; and*

(2) *All the heat rejected to the cold body be given it when the working material has the same temperature as that body.*

This is easily proved from the Tϕ-diagram, Fig. 21. Imagine heat to be received reversibly along some such line as $d'\ a'\ b$, with the temperature of the working substance varying from $T_{d'}$ to T_1. Obviously less heat is received than if the reception had been isothermal, because the area $fd'\ a'be$ $(= fabe - ad'a')$ is less than the area $fabe$. The work done is also less because the area $dd'a'bcd$ $(= dabc - ad'a')$ is less than $dabc$. But since the area $ad'a'$ is lost in both cases, and since the (smaller) area representing work is affected more than the (larger) area representing heat received, it follows that

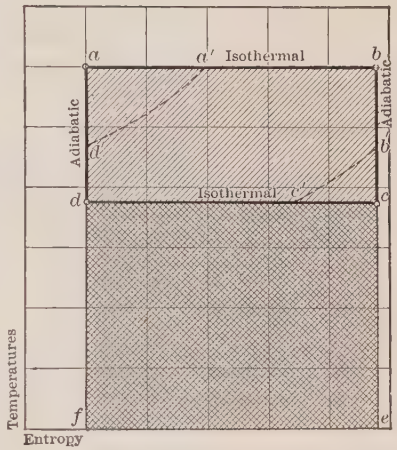


Fig. 21. — Tϕ-Diagram of Carnot Cycle.

$$\frac{a'bcdd'}{a'bcefd'} < \frac{abcd}{abef} \quad \text{and} \quad Ef' < Ef.$$

A similar proof would show that the rejection of heat along a line such as $b'c'$ also gives a cycle which is less efficient than that with isothermal heat rejection.*

* Confusion is sometimes caused by the apparent contradiction of these statements and those given as propositions (1) and (2) in Section 50. It should, however be noted that a very distinct limitation is put upon the reversible engines considered in that section. They receive *all* their heat at temperature T_1 from the hot body in reversible fashion and reject reversibly to a cold body at temperature T_2. On the other hand, engines receiving heat along such a line as $d'a'$, Fig. 21, could only do so reversibly by employing a string of hot bodies with temperatures

54. The Constant-Volume *Regenerative or Stirling Cycle.* (a) In this cycle, which is drawn to PV-coördinates in Fig. 22, the gas receives heat from the hot body and rejects heat to the cold body along the isothermals *ab* and *cd* exactly as in the case of the Carnot engine. The two adiabatics of this latter case are, however, replaced by the two constant volume lines *bc* and *da*.

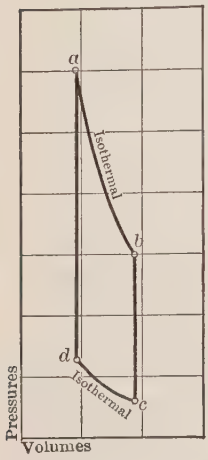


Fig. 22.—PV-Diagram of Constant-Volume Regenerative Cycle. Air as working substance. Same pressure range as in Fig. 18.

The line *bc* is supposed to be obtained by allowing the working substance to reject heat to a body so arranged that it stores that heat in such a manner that (1) each part of the body is always at the same temperature as the contiguous gas, (2) the temperature of each part remains constant, and (3) each increment of heat after storing is maintained at the temperature of reception. The line *da* is supposed to be obtained by the return of the stored heat to the gas by a reversal of this operation.

(b) Such a heat storing and restoring body is known as a **Regenerator** and in its perfect state is of course purely ideal. It may be approximated, however, by a long pipe of heat-insulating material filled with wire gauze or equivalent, and with temperature T_1 at one end and T_2 at the other. As hot gas flows through in the direction T_1T_2 it will impart heat to the walls and filling at a progressively decreasing temperature and give the change *bc*; while *da* may be obtained by causing gas to flow through the regenerator in the opposite direction.

The Mechanism of the Stirling Engine.

(c) The machinery necessary for the carrying out of such a cycle is shown in Fig. 23. The cylinders, Y and Y_1, and the hot and cold bodies, U and X, are similar to those used in the Carnot engine. The tube R is the regenerator just described and its

varying from T_d' to T_a' by *infinitesimal elements*, so that the gas might without sensible error be said to receive each element of heat when at the same temperature as the body supplying it. This is distinctly contrary to the assumptions of Section 50 as reiterated above.

contained volume is assumed to be negligible compared with that of either cylinder.

Imagine the piston in Y_1 at the bottom of the cylinder and that in Y at the top, as the result of the expansion *ab*, Fig. 22. Y is then filled with a gas with conditions shown at *b*. Now drive the right piston down, thus forcing the gas through the regenerator, and allow the left piston to rise at just the rate necessary to keep constant the *total volume of the gas*. During this process the regenerator will absorb heat and its temperature will grade from T_b at the right to T_c at the left. When the piston in Y has reached the bottom of its stroke all the gas will be in Y_1, the piston in the latter will be at the top of the stroke, and the constant-volume line *bc* will have been produced. Now hold the right piston stationary, bring the cold body up to cylinder Y_1 and force the piston into this cylinder, until the volume occupied by the gas is that shown at *d*, in Fig. 22. This will produce the isothermal compression. Free the piston in Y, continue the downward motion of that in Y_1 until it reaches the bottom of its cylinder, and simultaneously allow the right piston to rise as much as is necessary to keep the volume constant. This will give the line *da* of the diagram, and the gas in passing through the regenerator will rise in temperature from $T_d\,(=T_c)$ to $T_a\,(=T_b)$. Finally fix the left piston in its position, bring the hot body up to cylinder Y, and allow the gas to expand isothermally from *a* to *b*, completing the cycle.

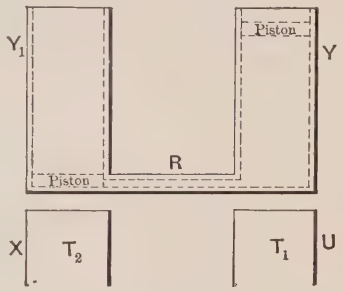


Fig. 23. — Machinery of Constant-Volume Regenerative Cycle.

Work Obtained per Unit Weight of Gas by Use of Constant-Volume Regenerative Cycle.

(d) The work theoretically available from an engine using this cycle can be found, as in the case of the Carnot engine (see Section 49 (d), by summing up the quantities of work done during each process of the cycle. This is done in the following tabulation in which the letters in the first column refer to Fig. 22. It is evident from the figure that, as before, the ratios of expansion and compression are equal. Thus, using *unit weight*,

Line.	Process.	Work Done $\left(\frac{\text{Ft.-Lbs.}}{\text{Lb.-Gas}}\right)$.
ab	Isothermal Expansion	$+ RT_1 \log_e r$
bc	Isovolumic Change	0
cd	Isothermal Compression	$- RT_2 \log_e r'$
da	Isovolumic Change	0

Net work per cycle $= RT_1 \log_e r - RT_2 \log_e r'$, and since $r = r'$,

$$= (T_1 - T_2)\, R \log_e r \text{ ft.-lbs.} \quad . \quad . \quad . \quad (73)$$

Thus

$$\Delta E = \frac{(T_1 - T_2)\, R \log_e r}{778} \text{ B.t.u.} \quad . \quad . \quad . \quad (74)$$

Efficiency of the Constant-Volume Regenerative Engine.

(e) *With an ideal regenerator the isovolumic processes bc and da would be thermodynamically reversible*, and the isothermal compression and expansion, as in the Carnot engine, are also reversible. *The cycle as a whole must then be reversible, and therefore its efficiency must equal the efficiency of the Carnot cycle.* This may be proved as follows:

(f) With *unit weight* of gas, the heat received from the hot body may be called ΔQ_1, and as before it must equal $\frac{RT_1 \log_e r}{778}$. The heat rejected to the cold body is similarly ΔQ_2 and is equal to $\frac{RT_2 \log_e r'}{778} = \frac{RT_2 \log_e r}{778}$, since r is equal to r'.

The external work done, or the mechanical energy made available, must equal the difference between the work done by the gas during the isothermal expansion and the work done upon it during the isothermal compression, hence it must be

$$778\, \Delta E = RT_1 \log_e r - RT_2 \log_e r.$$

Then the efficiency is

$$Ef = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1} = \frac{RT_1 \log_e r - RT_2 \log_e r}{RT_1 \log_e r} = \frac{T_1 - T_2}{T_1}. \quad (75)$$

There is often difficulty at first in realizing that this cycle, which has the same efficiency as the Carnot cycle, fulfills the criterion of maximum efficiency stated in Section 53. Careful study will show, however, that the statements in that section apply only to heat transfers between the working substance and

bodies external to the actual engine — i.e., the hot and cold bodies. The heat given up or received by the gas during the *constant-volume* changes, i.e., $C_v(T_1 - T_2)$, is really stored and restored reversibly and does not enter or leave the system.

Tϕ-Diagram of Constant-Volume Regenerative Cycle.

(g) Fig. 24 shows the Tϕ-diagram of this cycle as *abcd* superimposed upon that of the Carnot cycle *abc'd'*. For convenience

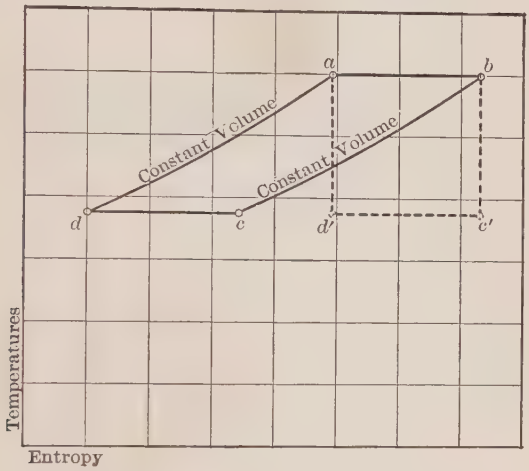


Fig. 24. — Tϕ-Diagram of Constant-Volume Regenerative Cycle. Same temperature range as in Figs. 18 and 21.

in comparison the two cycles are drawn for the same temperature range.

The lines *bc* and *da* are obtained from Eq. (63) and are evidently parallel curves. The areas *bc'c* and *ad'd* are, therefore, equal, hence *abcd* must equal *abc'd'*. Each of these areas, however, represents the heat converted into mechanical energy. The heat supplied in each case is shown by the area under *ab*. Therefore, the heat supplied in each cycle being the same, and the work done being the same, the efficiencies are equal.

55. The Constant-Pressure Regenerative, or Ericsson, Cycle. The PV-diagram for this cycle, shown in Fig. 25, differs from the cycle last considered only in the fact that the regenerator processes are carried on at constant pressure instead of at constant

volume. The same mechanism may be used as in the last case, and the cycle, being reversible, must have the same efficiency.

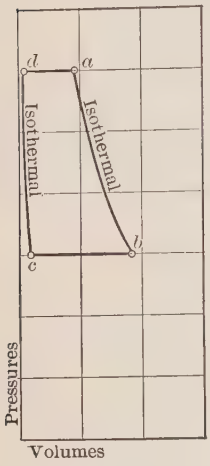


Fig. 25. — PV-Diagram of Constant Pressure Regenerative Cycle. Air as working substance. Same volume range as in Fig. 22.

The $T\phi$-diagram of this cycle is similar to that of the constant-volume cycle shown in Fig. 24. The curves corresponding to *bc* and *da* of that figure are obtained from Eq. (61) and of course have a different slope; otherwise nothing is altered, and statements concerning one of these cycles are, in general, true of both.

56. The Constant-Volume Heat-Change, Otto, or Beau de Rochas Cycle. (**a**) This cycle, the PV-diagram of which is shown in Fig. 26, consists of two adiabatics *bc* and *da* and two constant volume lines *ab* and *cd*. Heat is received from the hot body along the line *ab*, the pressure and temperature rising while the volume remains constant. Heat is rejected to the cold body along the line *cd*, the pressure and temperature dropping while the volume remains constant.

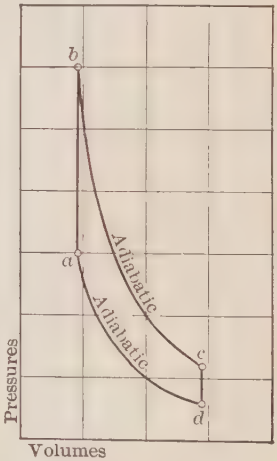


Fig. 26. — PV-Diagram of Otto Cycle. Air as working substance. Conditions at *b* same as those at *a* in Fig. 18 and lowest pressure same as in that case.

(**b**) The *reception of heat is irreversible*, since the temperature of the hot body is at least as high as that which the gas attains when reaching condition *b*, and therefore must be higher than that of the gas during the entire reception of heat ΔQ_1. The same is true for the rejection of heat along *cd*, the cold body having a temperature at least as low as that of the gas at *d*. This case is the first one cited in Section 35 as a process intrinsically irreversible.

This cycle is not only *irreversible*, but, as is evident, *it does not fulfill the criterion for maximum efficiency* (Section 53), and

hence has an efficiency lower than that of the cycles previously described. It is, however, the only one of the four gas cycles so far considered which is of any great practical importance.

Mechanical Energy Obtained per Unit Weight of Gas Operating in Otto Cycle.

(c) The following tabulation gives the mechanical energy changes for each line per *unit weight* of gas:

Line.	Type of Change.	Work in Ft.-lbs. Done by Gas.
ab	Constant-Volume Pressure Rise	0
bc	Adiabatic Expansion	$\frac{R(T_b - T_c)}{\gamma - 1}$
cd	Constant-Volume Pressure Drop	0
da	Adiabatic Compression	$-\frac{R(T_a - T_d)}{\gamma - 1}$

The summation of the last column gives the net work per cycle per unit weight of gas, as

$$\text{Net work} = \frac{R}{\gamma - 1}(T_b - T_c - T_a + T_d) \text{ ft.-lbs.} \quad . \quad (76)$$

From Eq. (33), $\frac{R}{\gamma - 1}$ is equal to K_v, giving

$$\text{Net work} = 778\,\Delta E = K_v(T_b - T_c - T_a + T_d). \quad . \quad (77)$$

(d) This same result could have been obtained more briefly as follows: The mechanical energy obtained must equal $\Delta Q_1 - \Delta Q_2$, when measured in heat units; that is, $\Delta E = \Delta Q_1 - \Delta Q_2$. Since the heat changes take place at constant volume, $\Delta Q_1 = C_v(T_b - T_a)$ and $\Delta Q_2 = C_v(T_c - T_d)$,
hence, in ft. lb. units,

$$\begin{aligned} 778\,\Delta E &= K_v(T_b - T_a) - K_v(T_c - T_d) \\ &= K_v(T_b - T_a - T_c + T_d) \end{aligned}$$

which is the same as Eq. (77).

Efficiency of Otto Cycle.

(e) Writing

$$Ef. = \frac{\Delta E}{\Delta Q_1} = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1}$$

and substituting in the last form gives

$$Ef. = \frac{C_v (T_b - T_a) - C_v (T_c - T_d)}{C_v (T_b - T_a)}$$

$$= 1 - \frac{T_c - T_d}{T_b - T_a}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

This expression can be further transformed and simplified so that important conclusions can be easily deduced. Since the curves *bc* and *da* are adiabatics, Eq. (51) gives

$$\frac{T_c}{T_b} = \left(\frac{V_b}{V_c}\right)^{\gamma-1} \quad \text{and} \quad \frac{T_d}{T_a} = \left(\frac{V_a}{V_d}\right)^{\gamma-1}$$

Since $V_a = V_b$ and $V_c = V_d$

$$\left(\frac{V_b}{V_c}\right)^{\gamma-1} = \left(\frac{V_a}{V_d}\right)^{\gamma-1}$$

and therefore

$$\frac{T_c}{T_b} = \frac{T_d}{T_a} \quad \text{and} \quad \frac{T_c - T_d}{T_b - T_a} = \frac{T_d}{T_a}. \quad . \quad . \quad . \quad (79)$$

Substituting from this in (78) gives

$$Ef. = 1 - \frac{T_d}{T_a} = \frac{T_a - T_d}{T_a}. \quad . \quad . \quad . \quad . \quad . \quad (80)$$

and

$$Ef. = 1 - \left(\frac{V_a}{V_d}\right)^{\gamma-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (81)$$

Thus it is evident —

(1) That *the efficiency* of this cycle *is independent of the upper temperature*, but *depends only upon the temperature range of adiabatic compression.*

(2) That with the same value of P_d, the *less the volume of one pound of gas at the end of compression the higher the efficiency.*

(3) That with the same temperature T_d, *the higher the temperature at the end of compression the higher the efficiency.*

Eq. (81) shows that *the efficiency of the cycle may vary with different real gaseous working substances* because the value of γ, as shown in Table I, is not the same for different gases. This is in marked contrast to the cycles previously considered, where the efficiency could be expressed entirely in terms of the tem-

peratures of the hot and cold bodies, and where the efficiency was independent of the individual characteristics of the gaseous working substances.

Writing the Carnot efficiency

$$\frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}$$

and the Otto efficiency

$$\frac{T_a - T_d}{T_a} = 1 - \frac{T_d}{T_a} = 1 - \frac{T_2}{T_a},$$

inspection shows that for the same upper and lower temperatures the Otto efficiency must be the smaller, as T_a must be less than T_1.

Tϕ -Diagram of Otto Cycle.

(f) In reality the Tϕ-diagram of this cycle cannot be drawn by the same means that was used in the preceding cases, for the reason that in Chapter VII *the equation* $\Delta\phi = \int \frac{dQ}{T}$ *was proved for reversible processes only*, whereas two processes in this cycle are irreversible (see (b)).

It is possible, however, to draw a Tϕ-diagram for this case by making use of the fact that *the entropy change accompanying an alteration from any given condition to another must always be the same, no matter how the change from the first state to the second occurs.*

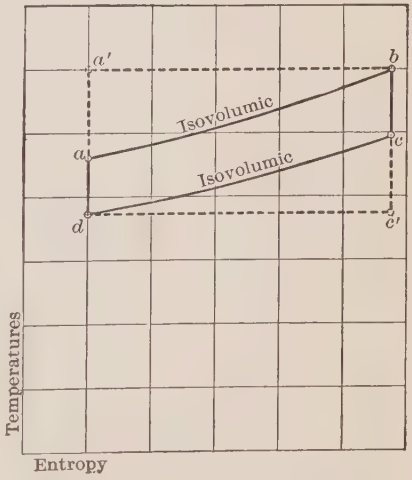


Fig. 27. — Tϕ-Diagram of Otto Cycle. Air as working substance. Same temperature range as that in Figs. 18 and 21.

To find the entropy changes occurring as the gas receives heat along the line *ab* in Fig. 27, it is then only necessary to discover some *reversible* method of supplying the same amount of heat in such a way that the condition of the gas at every individual point on *ab* will be the same as when the heat supply is irreversible.

Such a heating process would result from the use of a series of reservoirs with temperatures graded from T_a to T_b. The gas can then receive each increment of heat from a reservoir having the same temperature as it possesses at the instant, and therefore the gas can thus be heated reversibly.

The total entropy change would then be

$$\Delta\phi = C_v \log_e \frac{T_b}{T_a},$$

and by the use of this equation the Tϕ-diagram can be drawn as in Fig. 27. The diagram in this figure shows the same changes as are represented by PV-coördinates in Fig. 26. The dotted rectangle is the Carnot cycle originally given in Fig. 21.

57. The Constant-Pressure Heat-Addition, Brayton, or Joule Cycle. (a) This cycle, like the last, is an *irreversible* one in the thermodynamic sense, but it is important because of its practical application to certain purposes which will be considered later. It is now necessary to derive the type equations for the cycle, as has been done in the preceding cases.

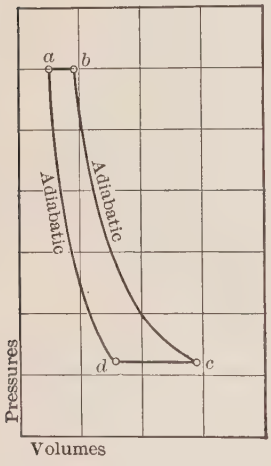


Fig. 28. — PV-Diagram of Joule Cycle. Air as working substance. Adiabatic expansion with same pressure range as in Fig. 26.

Fig. 28 shows the Joule cycle drawn to PV-coördinates. Starting at a, heat is added to the working substance by the hot body, volumes and temperature increasing at constant pressure, until the point b is reached. Obviously the temperature T_1, of the hot body, must be at least as high as that of the gas at b, and therefore higher than that of the gas at a. The heat addition is therefore irreversible.

From b the gas expands adiabatically to c, then rejects heat irreversibly, maintaining constant pressure until the volume V_d is reached, and is then compressed adiabatically to a, completing the cycle.

Mechanical Energy Obtained per Unit Weight of Gas Operating in Joule Cycle.

(b) As before, the useful effect per *unit weight* of gas can be found by tabulation. Thus:

Line.	Type.	Mechanical Energy (Ft.-lbs.) Made Available.
ab	Constant-Pressure Expansion	$+ P_a(\mathbf{V}_b - \mathbf{V}_a)$
bc	Adiabatic Expansion	$+\frac{P_b\mathbf{V}_b - P_c\mathbf{V}_c}{\gamma - 1}$
cd	Constant-Pressure Compression	$- P_c(\mathbf{V}_c - \mathbf{V}_d)$
da	Adiabatic Compression	$-\frac{P_a\mathbf{V}_a - P_d\mathbf{V}_d}{\gamma - 1}$

The summation of the last column gives for the cycle

$$\text{Net Work} = P_a(\mathbf{V}_b - \mathbf{V}_a) + \frac{P_b\mathbf{V}_b - P_c\mathbf{V}_c}{\gamma - 1} - P_c(\mathbf{V}_c - \mathbf{V}_d) - \frac{P_a\mathbf{V}_a - P_d\mathbf{V}_d}{\gamma - 1} \text{ ft.-lbs.} \quad (82)$$

(c) This expression could be simplified, but it is hardly worth while, as a shorter one can be obtained more easily in the following manner.

Writing available mechanical energy, or work done, as

$$\Delta E = (\Delta Q_1 - \Delta Q_2) \text{ B.t.u.},$$

it follows from the character of the lines *ab* and *cd* that

$$\Delta E = C_p(T_b - T_a) - C_p(T_c - T_d) = C_p(T_b - T_a - T_c + T_d) \text{ B.t.u.} \quad . \quad . \quad (83)$$

and

$$778\ \Delta E = K_p(T_b - T_a) - K_p(T_c - T_d) = K_p(T_b - T_a - T_c + T_d) \text{ ft.-lbs.} \quad . \quad . \quad (84)$$

Efficiency of Joule Cycle.

(d) Since

$$Ef. = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1}$$

it must be in this case,

$$Ef. = \frac{C_p(T_b - T_a) - C_p(T_c - T_d)}{C_p(T_b - T_a)}$$

$$= 1 - \frac{T_c - T_d}{T_b - T_a} \quad \ldots\ldots\ldots \quad (85)$$

This can be further simplified by using Eq. (52). From this

$$\frac{T_c}{T_b} = \left(\frac{P_c}{P_b}\right)^{\frac{\gamma-1}{\gamma}} \text{ and } \frac{T_d}{T_a} = \left(\frac{P_d}{P_a}\right)^{\frac{\gamma-1}{\gamma}}$$

then, since

$$P_a = P_b \text{ and } P_d = P_c,$$

$$\frac{T_c}{T_b} = \frac{T_d}{T_a} = \frac{T_c - T_d}{T_b - T_a}.$$

Substituting this in Eq. (85) gives

$$Ef. = 1 - \frac{T_d}{T_a}, \quad \ldots\ldots\ldots \quad (86)$$

a result similar to that obtained for the Otto cycle.

The last equation can, by simple substitution, also be written

$$Ef. = 1 - \left(\frac{V_a}{V_d}\right)^{\gamma-1} \quad \ldots\ldots \quad (87)$$

which is likewise similar to the corresponding form for the Otto cycle.

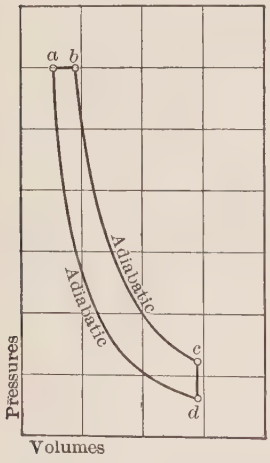


Fig. 29. — PV-Diagram of Diesel Cycle. Air as working substance. The lines *bc* and *cd* coincide with the lines similarly lettered in Fig. 26.

Tϕ-Diagram of Joule Cycle.

(e) By replacing the irreversible isobarics by equivalent reversible processes, the Tϕ-diagram to represent this cycle can be constructed, as was done for the Otto cycle; but as this diagram is of little practical value it will be omitted.

58. The Diesel Cycle. (a) This cycle, drawn to PV-coördinates, is shown in Fig. 29. The heat is added from the hot body

during the constant-pressure expansion *ab*, and then the gas expands adiabatically from *b* to *c*. Heat is discharged to the cold body while the pressure of the working substance decreases from *c* to *d* at constant volume. The cycle is closed by the adiabatic compression *da*. The Diesel cycle is *irreversible* for the same reasons that the Otto and Joule cycles are.

Mechanical Energy Obtained per Unit Weight of Gas Operating in Diesel Cycle.

(b) As before, the amount of mechanical energy made available can be found by tabulating:

Line.	Character.	Work (Ft.-Lbs.) Done by Unit Weight Gas.
ab	Constant-Pressure Expansion	$+ P_a (\mathbf{V}_b - \mathbf{V}_a)$
bc	Adiabatic Expansion	$+ \frac{P_b\mathbf{V}_b - P_c\mathbf{V}_c}{\gamma - 1}$
cd	Constant-Volume Pressure Drop	0
da	Adiabatic Compression	$- \frac{P_a\mathbf{V}_a - P_d\mathbf{V}_d}{\gamma - 1}$

The summation of the last column gives for the cycle

$$\text{Net Work} = 778\ \Delta E = P_a (\mathbf{V}_b - \mathbf{V}_a) + \frac{P_b\mathbf{V}_b - P_c\mathbf{V}_c}{\gamma - 1} - \frac{P_a\mathbf{V}_a - P_d\mathbf{V}_d}{\gamma - 1} \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

(c) This expression need not be simplified, since, as in previous cases, there is a more convenient way of finding a short expression for the work done. Writing

$$\Delta E = (\Delta Q_1 - \Delta Q_2) \text{ B.t.u.},$$

it follows that, in the case of the Diesel cycle,

$$\Delta E = C_p (T_b - T_a) - C_v (T_c - T_d) \text{ B.t.u.} \quad . \quad . \quad (89)$$

and

$$778\ \Delta E = K_p (T_b - T_a) - K_v (T_c - T_d) \text{ ft.-lbs.} \quad . \quad (90)$$

Efficiency of Diesel Cycle.

(d) Writing

$$Ef. = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1}$$

the efficiency in this case must be

$$Ef. = \frac{C_p (T_b - T_a) - C_v (T_c - T_d)}{C_p (T_b - T_a)}$$

$$= 1 - \left[\frac{C_v}{C_p} \cdot \frac{T_c - T_d}{T_b - T_a}\right] = 1 - \left[\frac{1}{\gamma} \cdot \frac{T_c - T_d}{T_b - T_a}\right] \quad . \quad (91)$$

This has the same form as Eq. (85), for the efficiency of the Joule cycle, with the exception of the introduction of $1/\gamma$. It should, however, be noted that the temperature term is not numerically the same in both cases, on account of the difference in the shape of the two cycles.

(e) By substituting reversible processes for the irreversible ones, a Tϕ-diagram equivalent to this cycle can be constructed.

GAS CYCLES — TABLE III

CYCLE		WORK–FT. LBS. PER LB. OF GAS.	EFFICIENCY
NAME	PV-DIAGRAM		
GENERAL	Any Number of Processes Enclosing an Area, A	= Area Enclosed (A) $=\Delta Q_1 \times$ Eff. $\times$ 778. $=778\ \Delta E$ $=778\,(\Delta Q_1 - \Delta Q_2)$	=Result ÷ Effort $=(\Delta Q_1 - \Delta Q_2) \div \Delta Q_1$
CARNOT	Isothermals and Adiabatics	$=(T_1 - T_2)\,R \log_e r$	$=\frac{T_1 - T_2}{T_1} = 1 - \left(\frac{T_2}{T_1}\right)$
STIRLING	Isothermals and Isovolumics	"	"
ERICSSON	Isothermals and Isobarics	"	"
OTTO	Adiabatics and Isovolumics	$=K_v(T_b - T_a - T_c + T_d)$	$=1-\left(\frac{T_d}{T_a}\right)=1-\left(\frac{V_a}{V_d}\right)^{\gamma-1}$
BRAYTON OR JOULE	Adiabatics and Isobarics	$=K_p(T_b - T_a - T_c + T_d)$	"
DIESEL	Adiabatics Isobaric and Isovolumic	$=K_p(T_b - T_a) - K_v(T_c - T_d)$	$=1-\left[\frac{1}{\gamma} \cdot \frac{T_c - T_d}{T_b - T_a}\right]$

CHAPTER IX.

VAPORS.

59. Vapors and Gases. When materials change from the liquid* to the gaseous state they do not immediately reach the condition in which their behavior even approximately obeys the laws of ideal gases. It is customary to designate materials as **Vapors** when in this intermediate condition. It will appear later that when strictly interpreted the term vapor will apply to many of the materials with which the engineer deals and which he is accustomed to call gases.

60. Formation of Vapor. (a) When a liquid is heated under constant pressure its temperature will first rise until it reaches a certain *temperature which is dependent upon the pressure under which it exists;* after which further *addition* of heat will cause some of the material to *change physical state at constant temperature,* this temperature being the one fixed by the pressure existing. The amount of material that has changed state will increase as this further addition of heat progresses, and if sufficient heat is added all the liquid present will thus change its state. The material formed during this change of state is called a **vapor.**

(b) Considering the process for the first time, one would recognize two possible methods of formation of vapors, and without previous knowledge would not be able to decide between them. Thus:

(1) The liquid as a whole might gradually change from liquid to vapor, all of it being at any one time in exactly the same condition of transformation. Or,

(2) *Parts* of the liquid might *progressively* change to vapor as the *necessary heat* became *available,* leaving the remainder still in the form of liquid.

Usually vaporization occurs by method (2), and as heat is added more and more vapor appears at the expense of liquid.

* Or directly from the solid, as in "sublimation."

Thus when one-fourth of the total heat necessary for complete vaporization is added one-fourth of the liquid will be vaporized, and so on until vaporization is complete.

(c) In the sections which follow the generation of vapor may be conveniently studied by imagining the process carried out in the device illustrated in Fig. 30. It consists of a vertical cylinder with closed end down, containing a frictionless piston of given weight, — all being placed under a bell jar in which a perfect vacuum is maintained.

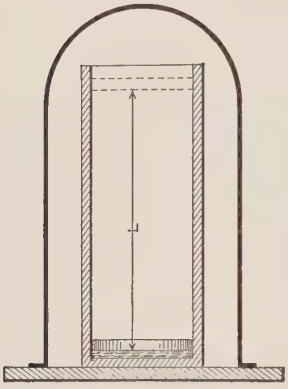


Fig. 30.

Assume now that one pound of liquid is inclosed in the cylinder beneath the piston. The total pressure on the upper surface of this liquid will be that due to the weight of the piston, and since it is evenly distributed over the entire surface it may be designated as P pounds per square foot of surface.

Any liquid may be used and, in general, may have any temperature between that of solidification and that of vaporization at the chosen pressure. It is, however, customary to assume the temperature at a convenient value dependent on the physical characteristics of the liquid dealt with.

In the case of water, and of all other liquids for which such a temperature is at all convenient, the engineer is accustomed to refer all vaporization phenomena to a datum temperature of 32° F. As this is the melting temperature of ice under ordinary conditions, it is readily checked and is hence a very satisfactory standard.

To make the results of the process under consideration conform to the engineering reality, the liquid beneath the piston, in Fig. 30, will be assumed at 32° F.

61. Heat of the Liquid. (a) If heat is added to the liquid beneath the piston in Fig. 30, the temperature will rise and, in the case of water, at the approximate rate of 1° F. for each B.t.u., since the specific heat of water at constant pressure is approximately 1. In any case the rise will take place at the rate of

1° F. for each addition of heat equal to C_p, the constant-pressure specific heat of the liquid dealt with. This will continue until a temperature is reached at which vaporization begins. This temperature will depend upon the value of the pressure, and in any case has to be determined by experiment. Thus with water at atmospheric pressure (equal to 14.7 pounds per square inch, or $14.7 \times 144 = 2116.8$ pounds per square foot), the temperature will be 212° F.; while for a pressure of 100 pounds per square inch (equal to $100 \times 144 = 14{,}400$ pounds per square foot) the temperature will be about 327° F. These various temperatures are called the **Temperatures of Vaporization** and will be designated by the symbols t_v and T_v respectively for Fahr. and Absolute temperatures. When it is necessary to indicate a particular temperature, the corresponding pressure in pounds per square inch will follow the subscript v; thus, the temperature Fahr. of vaporization at atmospheric pressure would be $t_{v14.7}$ or temperature absolute $T_{v14.7}$.

(**b**) The heat added during the process of raising the temperature from 32°, or other datum level, to the temperature of vaporization is called the **Heat of the Liquid** and is designated by q. Obviously it has a different value for every different pressure and it is customary to tabulate these values with others in so-called **Vapor Tables.** In general

$$q = \int C_p \, dT, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

the integration being performed between the datum temperature as the lower limit and the temperature corresponding to the pressure in question as the upper limit.

If the specific heat of water were exactly equal to unity at all temperatures, the value of q for this material for any temperature or pressure of vaporization could be found from the equation

$$q = t_v - 32; \quad . \quad . \quad . \quad . \quad . \quad . \quad (93)$$

and since these values vary but slightly from those determined by experiment, this equation is often used by engineers. For accurate work the experimentally determined values given in the steam tables should be used.

(**c**) Eq. (93) could not be used, even as an approximation, with any liquid other than water, since it depends upon the assumption that the specific heat of the liquid is invariably

equal to unity. If, as before, the specific heats of liquids at constant pressure are designated by C_p, and if they are assumed constant over the ranges of temperature considered, the equation

$$q = C_p (t_v - 32) \quad . \quad . \quad . \quad . \quad . \quad . \quad (94)$$

may be used in determining the heat of the liquid for any temperature or pressure of vaporization. Note that there are liquids which vaporize at ordinary pressures below the temperature of 32° F. In such cases a datum temperature lower than 32° may be taken from which the heat of the liquid is calculated. This necessitates a different form of equation. In its most general expression this would become

$$q = C_p (T_v - T_0), \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

where T_0 stands for any arbitrarily chosen datum.

62. Latent Heat of Vaporization. (a) Consider now the pound of liquid which has been raised to the temperature t_v. With further addition of heat vaporization occurs. The marked characteristics of vaporization under the assumed conditions are (1) the very great *increase of volume at constant temperature and pressure*, (2) the *change of the physical state* of the material from liquid to vapor, and (3) the enormous *quantity of heat absorbed*.

(b) The process carried out in the apparatus of Fig. 30 would result in driving up the piston to some higher position in the cylinder, against the pressure exerted by that piston on the upper surface of the vapor. Evidently, here, force would act through distance and therefore external work would be done. This work could not be done without a supply of energy, and, since heat energy is the only form supplied during the process, it follows that at least some of this heat must have been used for the doing of the external work. Let F be used to designate the area of the piston face in square feet, and L the number of feet the piston is moved during the vaporization of the entire pound of liquid under consideration.

Then the foot-pounds of external work done per unit weight are

$$778\,\Delta E = (PF)\,L,$$

which, rearranged, becomes

$$778\,\Delta E = P\,(FL) = P\,(V_2 - V_1), \quad . \quad . \quad . \quad (96)$$

where $\mathbf{V}_1$ represents the volume occupied by the liquid and $\mathbf{V}_2$ that occupied by the vapor. It is customary to designate the *increase of volume* ($\mathbf{V}_2 - \mathbf{V}_1$) by the letter u, hence the external work done, in *foot-pounds*, is

$$778\,\Delta E = Pu, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (96a)$$

and its value in thermal units can be found by dividing Pu by 778. Representing $\frac{1}{778}$ by A, the expression for the B.t.u. of heat used in the doing of external work becomes

$$\Delta E = APu \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (97)$$

This quantity is called the **External Latent Heat of Vaporization.** It has a different value for every different pressure at which vaporization takes place, and these values are tabulated in the Vapor Tables already mentioned.

It is very necessary to observe that the term external "*latent heat*" is a misnomer. The heat used for the doing of external work does not exist as heat energy *in* the vapor, for, during the process of vaporization, it is changed into mechanical energy which is extraneous to the vapor itself. The case is somewhat similar to isothermal expansion of a gas. This heat is converted into mechanical energy as rapidly as received. Hence, in a piston engine, the external latent heat may be considered as external work delivered by the piston rod. If the energy after reception can be said to be "latent," it must be latent *mechanical* energy and not latent heat energy. It is stored, if stored at all, in the piston or other similar part of the apparatus, and is in no sense *in* the vapor.

(c) Experiment shows that the heat used during the process of vaporization is not all accounted for by the external latent heat. Inspection of Eq. (1),

$$\Delta Q = \Delta S + \Delta I + \Delta E,$$

suggests the reason. In this case ΔQ represents the heat added to vaporize the liquid. As the temperature does not change during vaporization, no heat can be used as sensible heat, hence $\Delta S = 0$; but some of it may be used for the doing of internal work, ΔI. In fact the striking change of properties during this process could not occur without a very great readjustment within the material. The part of ΔQ which does not become

external latent heat is supposed to be used for doing this internal work, and is therefore called the **Internal Latent Heat.** It is designated by the symbol ρ and is tabulated in the Vapor Tables.

Recent work has led to the conclusion that liquid water is a more complex material than was originally supposed. It seems probable that instead of being simply a collection of molecules with formula H_2O it is really a mixture of at least three different kinds of molecules, H_2O, $(H_2O)_2$ and $(H_2O)_3$. It also seems probable that during the formation of vapor some of the more complex molecules break up into simpler form. If this is so, a possible use of at least part of the Internal Latent Heat in the case of water becomes evident since it would be used for breaking up the complex molecules. "Internal" latent heat would then be a correct name to apply to this part of the heat as it is latent *within* the substance, though there is room for argument as to whether it is latent as heat.

(d) The sum of the two latent heats, ρ and APu, is called the **Total Latent Heat of Vaporization,** and is designated in the tables by r. Thus $r = \rho + APu$

63. Total Heat per Pound of Vapor. (a) Using symbols, the total heat, above the *arbitrarily chosen datum temperature, per pound* of vapor at any pressure P, is the sum of the sensible heat, the internal latent heat, and the external latent heat; thus it is

$$q_p + \rho_p + (APu)_p = q_p + r_p, \quad . \quad . \quad . \quad . \quad (98)$$

and calling this λ gives

$$\lambda_p = q_p + r_p, \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

which is also given in the tables.

(b) Had the addition of heat in the process under consideration ceased before the entire pound of liquid had been vaporized, the cylinder would have contained both vapor and liquid at the same temperature. Representing by y the fraction of the total pound vaporized, the "heat of the vapor" * present must be

$$\Delta Q' = yq_p + y\rho_p + y\,(APu)_p$$

* The expression "heat of" will hereafter be used to designate the quantity necessary to bring the material in question to the condition under consideration, either from liquid at datum temperature or from liquid at the temperature of vaporization. The context will indicate which is referred to in any case.

and that of the remaining liquid must be

$$\Delta Q'' = (1 - y)\, q_p,$$

hence the total heat of the material in the cylinder is

$$\Delta Q_{yp} = \Delta Q' + \Delta Q''$$

$$= q_p + y\rho_p + y\,(APu)_p \quad . \quad . \quad . \quad . \quad . \quad (100)$$

$$= q_p + yr_p, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (100a)$$

which will be equal to Eq. (98) when $y = 1$, that is, when the entire pound has been vaporized.

64. **Saturated Vapor.** (a) The process assumed in the previous sections is really more or less idealized. In real cases, such as that taking place in the steam boiler, the vaporization does not progress so quiescently that the vapor separates entirely from the liquid and collects above it in the simple fashion already described. Instead, the formation of vapor is generally more or less violent, and, in separating from the body of the liquid, the vapor carries with it small drops of that liquid still unvaporized but mechanically entrained. These may often be carried great distances by a stream of vapor, and their separation from that vapor frequently presents considerable difficulty.

(b) Such mixtures of vapors and liquids are called **Wet Vapors,** to indicate the presence of the liquid; and when the entrained moisture has been entirely eliminated the material is called **Dry Vapor.** Since, under the conditions assumed in connection with Fig. 30, the liquid must all be raised to the temperature of vaporization before any of it can be converted into vapor at the same temperature, it follows that the **vapor** and **liquid** in such a wet mixture are in **thermal equilibrium;** that is, if there is any tendency for heat transfer from liquid to vapor, there is an equal tendency towards transfer in the opposite direction. With no heat lost to surrounding materials, such a mixture would maintain a constant composition indefinitely.

Vapor when in thermal equilibrium with its liquid is called **Saturated Vapor.** It is termed *Wet Saturated Vapor,* or simply *Wet Vapor,* if containing entrained liquid and *Dry and Saturated Vapor,* or simply *Dry Saturated Vapor,* if free from moisture in suspension.

(c) At different pressures the quantity of heat necessary to maintain material in the condition of dry saturated vapor has different values, being greater the higher the pressure. Abstraction of heat without change of pressure (and therefore without change of temperature) will cause partial or total condensation, but any vapor remaining will still be saturated vapor exactly like that which existed before condensation occurred. Therefore saturated vapor may be described as vapor so near the point of liquefaction that the removal of the slightest quantity of heat will produce partial condensation. Or (see following paragraphs) it may be described as *vapor in which the maximum number of molecules*, consistent with the maintenance of a vaporous state at the given pressure, *exist in a given space.*

65. Quality. (a) Practically all saturated vapors in actual use contain some entrained moisture, and it is often necessary to express just how much of each pound of such a mixture is liquid and how much is vapor. This is done by using the fraction representing the proportion of mixture which is really saturated vapor. This fraction is denoted by x, and is called the **Quality Factor,** or **Quality** of the vapor or mixture.

Thus if x is $\frac{3}{4}$, or 75 per cent, it means that three-quarters of every pound of mixture is vapor and the other quarter is liquid. The quality of the mixture would then be said to be 75 per cent.

(b) The heat content above datum temperature of such a mixture could obviously be found by putting x in place of y in Eq. (100), since, so far as associated heat is concerned, it makes no difference whether the vapor and liquid are separated or intimately mixed. For wet vapor of quality x, the total heat above datum temperature is then

$$\Delta Q_{xp} = q_p + x\rho_p + x\,(APu)_p = q_p + x\,r_p. \quad . \quad (101)$$

66. Superheated Vapor. (a) Having converted an entire pound of liquid into dry and saturated vapor in the apparatus of Fig. 30, its condition may be further modified if the addition of heat is still continued. Experiment shows that this further addition causes the temperature of the vapor to rise above that which existed during vaporization. This process is known as superheating, that is, raising above the saturation temperature corresponding to the existing pressure. The material formed is

called **Superheated Vapor,** and it becomes more and more like an ideal gas as its temperature is raised at constant pressure. Thus *it increases in volume with the addition of heat,* and a *given space must hold fewer and fewer molecules as the rise of temperature continues.*

(b) To make the meaning of the term "*saturated*" clearer, imagine a superheated vapor to be cooled, at constant pressure, by the removal of heat. As temperature decreases the volume also becomes less, and any given space holds more and more molecules until the temperature of vaporization is reached, at which point the material is reduced to the saturated condition. There is then, in a given space, the maximum number of molecules which can exist as vapor under the conditions obtaining; and further removal of heat would allow some of these to collect and form molecules of liquid, — that is, it would cause partial condensation. The material remaining uncondensed would still be saturated vapor, and with further removal of heat more and more of it would condense until finally all would become liquid, if the removal of heat were continued sufficiently far.

67. Heat per Pound of Superheated Vapor. The amount of heat added during superheating at constant pressure, to any temperature T_s, as described above, depends upon two things, — on the degree of superheat, which will be called D and equals $(T_s - T_v)$, and on the specific heat C_p of the vapor. Then the heat added during superheating would be given by the following equation if C_p happened to be a constant:

$$\Delta Q_D = C_p D. \quad . \quad . \quad . \quad . \quad . \quad . \quad (102)$$

The total heat (above datum temperature) of one pound of superheated * vapor would be

$$\Delta Q_s = q_p + \rho_p + (APu)_p + C_p D \quad . \quad . \quad . \quad (103)$$

$$= q_p + r_p + C_p D = \lambda_p + C_p D. \quad . \quad . \quad (103a)$$

* Recent experiment has shown that liquid water can exist for a considerable length of time within a mass of superheated steam, despite the fact that the two are not in thermal equilibrium. This fact must sometimes be taken into account in dealing with superheated steam in practical problems, when sufficient time does not elapse to establish thermal equilibrium. The heat per pound of such a mixture would be

$$Q_{xs} = q_p + xr_p + x\bar{C}_p D.$$

68. Diagram of Heat Changes during Vaporization. (a) The heat changes associated with the process of vaporization can all be graphically represented, as in Fig. 31, by plotting temperature as ordinates and heat added as abscissas. The figure is for water, but a similar diagram could be drawn for any material whose physical constants are sufficiently well known.

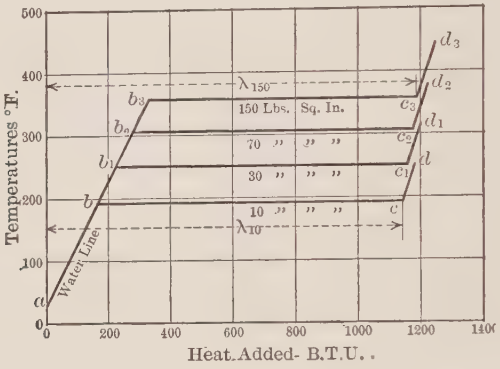


Fig. 31. — TQ-Diagram for Vapor Phenomena.

The line *abcd* shows the relation of temperature to heat added while one pound of water under 10 pounds pressure is first heated from 32° F. to vaporization temperature (line *ab*), is then completely vaporized (line *bc*), and finally is superheated through a limited range (line *cd*). The lines $ab_1c_1d_1$, $ab_2c_2d_2$, etc., show the same things for the other pressures indicated.

(b) A diagram drawn to a sufficiently large scale would show the line abb_3, and lines *cd*, c_1d_1, etc., as slightly curved because of the variation in the value of the specific heat of liquid water and of the specific heat, C_p, of superheated water vapor. In drawing Fig. 31 an average specific heat was used for the liquid and an average over each temperature range *cd*, c_1d_1, etc., for the superheated vapor. The latter accounts for the slight differences in slope of the superheating lines.

(c) The diagram shows how great an amount of heat is absorbed during the process of vaporization as compared with that used in bringing the liquid to the temperature of vaporization, or with that used in superheating. This is of great importance in heat engineering and will be fully considered later.

Two other facts of importance are made evident by the dia-

gram: One is the small change of total heat, λ, for a wide pressure range, as is seen by comparing the abscissas of c, c_1, etc.; and the other is the decrease of the total latent heat of vaporization, r, as the pressure rises.

(d) This figure also shows the temperature changes and heat given up when superheated vapor at any of the given pressures is cooled to the saturated condition, then is condensed, and the resulting liquid cooled to 32° F. The engineer must often consider changes in this direction.

69. Vapor Tables. Since the various values of q, ρ, APu, r, and λ are very frequently used by engineers and scientists, they are recorded, as already intimated, in the so-called Vapor Tables. There is of course a table for each material dealt with, so that it is customary to speak of "Steam Tables," "Ammonia Tables," "Carbon Dioxide Tables," etc.

The various values of each quantity are usually tabulated in vertical columns, the first two columns giving pressures and corresponding temperatures of vaporization, and the succeeding columns giving the corresponding values of the various heat quantities. Certain other columns are usually added containing such values as the volume occupied by a pound of liquid and by a pound of dry and saturated vapor. See tables in Appendix.

70. Saturation Curve. (a) Experiment shows that just as the saturated vapor of a given material at any particular temperature always exerts the same definite pressure, so also does one pound of dry saturated vapor at any temperature always occupy a definite volume. This latter is called the **Specific Volume** and is tabulated in the vapor tables. If the specific volumes are plotted against the corresponding pressures, the locus of the points is a PV-diagram similar to Fig. 32, which like the last is drawn for water vapor.

(b) This curve, called the **Saturation Curve,** may be very useful. If one pound of material at a given pressure has a volume represented by a point which falls to the left of the saturation curve, the material must be wet vapor; but if the point falls to the right of that curve, the material must be superheated vapor.

In the case of most engineering materials, the volume occupied by one pound of liquid is negligible as compared with that of one pound of vapor. In the case of water, the volume increases

nearly 1700 times when changing from liquid to dry saturated vapor under atmospheric pressure. If the volume of the liquid present be neglected, steam of 50 per cent quality would occupy 0.5 the volume it would if dry and saturated, and steam of 75 per cent quality would have 0.75 of the volume of dry saturated steam, and so on.

It follows that, if one pound of mixture is found to occupy a volume *ab*, Fig. 32, at the pressure indicated, it must have a

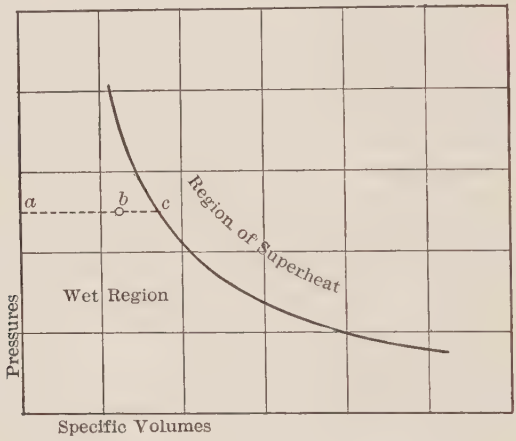


Fig. 32. — Saturation Curve for Water Vapor.

quality of $x = \frac{ab}{ac}$, if the volume of the water present is neglected. The case of superheated steam will be considered later, after the discussion of the experimental results.

The area to the left of the saturation curve might be called the *region of wet saturated vapor;* and the area to the right, the *region of superheated vapor.* The curve itself would then represent the boundary between the two, thus emphasizing the fact that dry saturated vapor is a unique condition at each pressure.

(c) Because of the resemblance of the saturation curve to an expansion curve there is a tendency to regard it as representing a possible expansion of vapor, that is, as an increase of volume during which the vapor remains dry and saturated throughout the entire process. Such an expansion might be obtained under very forced conditions, but normally no such process could be made to occur. It is then best to regard this

curve only as a boundary line between two fields and not as the graph of a process.

71. Defining Conditions for Saturated Vapors. In dealing with ideal *gases* the variables to be considered are pressure, temperature, and volume. They are so interrelated that fixing any *two* determines the third.

In the case of *dry saturated vapors*, however, the pressure, temperature, and volume are so related that the fixing of *one* determines the other two. This is not true of wet saturated vapors nor of superheated vapors.

In the case of *wet saturated vapors*, the fixing of temperature determines the pressure, and *vice versa*, but the *quality must be known in order to determine the volume occupied.*

Superheated vapors are more or less like gases, and in general the fixing of any two of the variables determines the third.

72. Evaporation. (a) There is sometimes difficulty in harmonizing the phenomena of vaporization, just described, with what is commonly known as evaporation. There is no real difference in the phenomena, vaporization as so far considered being only a *limiting case* of evaporation.

(b) In what follows it will be of material assistance if it is remembered that the so-called temperature of vaporization at any pressure is really the temperature of saturated vapor (wet or dry) at that pressure.

Thus the pressure exerted by a saturated vapor is determined by the temperature of the space the vapor occupies, and the pressure corresponding to any temperature can be found in the vapor table for the material.

(c) *Experiment shows that when the surface of a liquid is exposed to a space which is not already filled with the saturated vapor of that liquid, vapor is generated until the space is filled with such saturated vapor,* unless the liquid present is insufficient in amount. Of course vaporization ceases if the liquid is exhausted.

If the condition of equilibrium is reached, the saturated vapor must exert the pressure corresponding to the temperature of the space occupied. Until this equilibrium is attained, any vapor present must be superheated vapor because the number of molecules in a given space is less than would be the case if the space were filled with saturated vapor. Superheated vapor, however,

exerts a pressure less than that exerted by saturated vapor at the same temperature.

It follows that the pressure under which the liquid changes to vapor must constantly increase until a maximum is reached, when the space becomes filled with saturated vapor. After that, there can be no further change in the relative quantities of liquid and vapor present unless temperature changes.

(d) Since heat is required to change a liquid to a vapor, a supply of heat must be obtained from some source to cause "evaporation." If heat is not supplied from external sources, it is taken from the liquid and surrounding matter; hence the sensation of cold when alcohol, or other volatile liquid, is quickly evaporated from the skin.

The actual amount of heat necessary for evaporation may be found by considering the process *after equilibrium is attained.* Every pound of dry saturated vapor must have associated with it the total heat λ corresponding to the existing pressure and temperature.

(e) Usually the space into which the vapor passes contains other material beside the vapor; for example, some air is almost always present. **Dalton's law** states that *each constituent of such mixtures behaves as though the others were not present.* Therefore, the phenomenon is not in any way complicated by the presence of any number of other vapors and gases. The evaporation goes on until the space is filled with the saturated vapor of the liquid in question, and only then is equilibrium reached. The vapor will then have all the properties given numerically in its vapor table opposite the existing temperature.

The reason for calling vaporization as first considered a limiting case of what is generally known as evaporation should now be evident. The apparatus used in explanation was so arranged that the space available automatically increased as saturated vapor became available to fill it. This was done for simplicity and because of the close resemblance to the process taking place in the steam boiler, from which the vapor is withdrawn as rapidly as it is generated.

Note that the final conditions are the same in either case. A certain space is filled with saturated vapor of a given material, and what is true of that vapor in one case is true in the other.

When a space is thus filled with the saturated vapor of a

material, it is said to be *saturated with that vapor* or *with respect to that vapor*. Because of a peculiar construction of this expression an incorrect idea has become fixed in engineering language. It is usual to speak of air saturated with water vapor, whereas the real meaning is that a space occupied by air is *also* occupied by saturated water vapor.

(f) **Dalton's law** is sometimes called the **Law of Partial Pressures.** From the previous statement of this law it is evident (1) *that when several gases and vapors occupy a space in common, each behaves as though the others were absent, and* (2) *the pressures upon the walls enclosing the space, or at any point within the space, must be the sum of the pressures* exerted by all the constituents of the mixture. This pressure is called the **total pressure** of the mixture, while the pressures due to each of the constituents are called **partial pressures.**

If each constituent may be considered as obeying the laws of ideal gases, the same is true of the mixture. The pressure in the vessel, then, would be the total pressure, the volume would be that occupied by the mixture, and the temperature would be that of the mixture, which temperature must be the same for all constituents.

When some of the constituents of such a mixture are saturated vapors, the perfect gas laws cannot ordinarily be used if great accuracy is desired. When, however, the quantity of such vapors is small as compared with that of the gases present, the error resulting from the use of the gas laws is small, and for the sake of simplicity those laws are generally used and the error is neglected.

73. Boiling. Heat is often added to a liquid at such a rate and in such a way that the temperature of one part becomes higher than the temperature of adjacent parts; that is, local heating takes place. This is the result when the local addition of heat exceeds the rate of heat conduction through the material. Such heating raises the temperature locally to that of vaporization corresponding to the pressure, after which further addition of heat would cause local vaporization; that is, a small amount of the liquid inclosed within the rest would be converted into vapor and appear as a bubble.

The pressure at any point within a liquid at rest must be that

due to the *static head* of the liquid above that point *plus the pressure due to any material resting upon the surface.* Therefore, the bubble of vapor would be formed under that pressure and, during formation, would have to displace the column, or " piston," of water above it against that pressure.

The bubble, being less dense than the surrounding liquid, would rise, but if the temperature of the liquid encountered was lower than its own it might entirely condense before reaching the surface. This process continued long enough would bring all the liquid approximately to the same temperature, after which the vapor bubbles could travel upward through the liquid and escape as vapor from the surface.

Liquid is said to be in a **state of ebullition** or to be **boiling** when it is in such a state that bubbles of vapor formed within its mass pass up and out through its surface.

From what has preceded it can be seen that this process will occur when the body of water is at such a temperature that the pressure of its saturated vapor is equal to that upon its surface. This is sometimes given as a definition of boiling temperature.

74. Temperature-Entropy Changes of Vapors. (a) All the processes described in connection with the formation of vapor are thermodynamically *reversible;* hence for vapors, just as was done for gases in Section 38 (a), dE may be substituted for $APdV$ in the general Eq. (53) defining an infinitesimal entropy change. Then for such a change in a *unit* weight of vapor the expression becomes

$$d\phi = \frac{dS + dI + dE}{T}, \quad . \quad . \quad . \quad . \quad . \quad (104)$$

or

$$d\phi = \frac{dQ}{T}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (105)$$

and for a finite change

$$\Delta\phi = \int_1^2 \frac{dQ}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad (106)$$

These expressions may be used for determining the entropy changes for unit weight of any vapor when undergoing any *reversible* processes.

(b) The reversible temperature-entropy changes occurring during the vaporization of water at several different pressures are shown graphically in the Tϕ-diagram given in Fig. 33.

During the heating of the liquid at constant pressure the specific heat C_p, or heat required per pound per degree, may be either variable or constant. The equation for the lines ab, ab_1, ab_2, etc., for the entropy change experienced by the liquid,

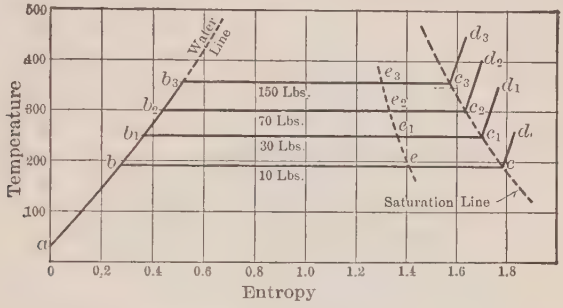


Fig. 33. — Tϕ-Diagram for Water and Water Vapor.

called briefly the **entropy of the liquid,** must be the same as Eq. (61) and is

$$\Delta\phi_l = \int_1^2 \frac{C_p\,dT}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad (107)$$

or

$$\Delta\phi_l = \overline{C}_p \log_e{}^* \frac{T_2}{T_1}. \quad . \quad . \quad . \quad . \quad . \quad (108)$$

This last equation can be used even if the specific heat is not a constant, by interpreting $\overline{C}_p$ as the mean value over the given temperature range.

(c) The process of vaporization is a constant-temperature or isothermal one; here, following Eq. (65), the entropy change experienced by the material during vaporization, called briefly the **entropy of vaporization**, is evidently

$$\Delta\phi_v = \frac{\Delta Q_v}{T_v} = \frac{r}{T_v}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (109)$$

where T_v is the temperature of vaporization.

(d) During superheating of the vapor at constant pressure the specific heat may be either variable or constant, and, paralleling Eq. (61), the entropy change, called briefly **entropy of superheating, is**

$$\Delta\phi_D = \int_{T_v}^{T_s} \frac{C_p\,dT}{T}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (110)$$

* It is usually more convenient to use $\log_{10}$ instead of $\log_e$. Since $\log_e = 2.302 \log_{10}$, Eq. (108) may be written $\Delta\phi_l = \overline{C}_p \times 2.302 \log_{10} (T_2/T_1)$. The other logarithmic equations which are to follow may be similarly transformed.

or $$\Delta\phi_D = \overline{C}_p \log_e \frac{T_s}{T_v} = \overline{C}_p \log_e \frac{T_v + D}{T_v}, \quad . \quad . \quad . \quad (110a)$$

where $\overline{C}_p$ is the mean specific heat, D is the temperature increase above the saturation temperature, T_v, and $T_s = T_v + D$.

(e) Summing up these results gives the total entropy change experienced by the material when transformed at constant pressure from liquid at datum temperature to superheated vapor at temperature $(t_v + D)$. This, which is briefly called the **total entropy of superheated vapor,** is

$$\Delta\phi_s = \Delta\phi_l + \Delta\phi_v + \Delta\phi_D. \quad . \quad . \quad . \quad . \quad . \quad (111)$$

Then $$\Delta\phi_s = \int_{T_0}^{T_v} \frac{C_p\, dT}{T} + \frac{r}{T_v} + \int_{T_v}^{T_s} \frac{C_p\, dT}{T} \quad . \quad . \quad (112)$$

$$= \overline{C}_p \log_e \frac{T_v}{T_0} + \frac{r}{T_v} + \overline{C}_p \log_e \frac{T_v + D}{T_v}, \quad . \quad (112a)$$

in which T_0 is the datum temperature, T_v is the temperature of saturation, and $T_s\,(= T_v + D)$ is that of the superheated steam when the amount of superheat is D degrees.

(f) If vaporization ceases before the entire pound of material has been vaporized, only a part, xr, of the total latent heat of vaporization, r, will have been added. The entropy change experienced by the material in coming to the condition of **wet saturated vapor** with quality x would then be

$$\Delta\phi_x = \Delta\phi_l + x\Delta\phi_v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (113)$$

$$= \overline{C}_p \log_e \frac{T_v}{T_0} + \frac{xr}{T_v}. \quad . \quad . \quad . \quad . \quad . \quad (114)$$

When x becomes unity, — that is, when vaporization is just complete, — there is **dry saturated vapor,** and this equation becomes

$$\Delta\phi_{sa} = \Delta\phi_l + \Delta\phi_v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (115)$$

$$= \overline{C}_p \log_e \frac{T_v}{T_0} + \frac{r}{T_v}. \quad . \quad . \quad . \quad . \quad . \quad (116)$$

The points e, e_1, e_2, etc., in Fig. 33, show the entropy change for different pressures as determined by Eq. (114) when $x = 0.75$. Obviously the distances be, b_1e_1, etc., must be 0.75 of the distances bc, b_1c_1, etc. This diagram then furnishes a means of determining quality in a manner similar to that used in the case of the saturation curve, Fig. 32, but is not subject to the approximation there necessary.

75. Continuity of the Liquid and Gaseous States. (a) It has been stated, in Chapter IV, that no real gases obey exactly the laws of ideal ones, but that it may be assumed without great error that those real gases which are farthest removed from the conditions of liquefaction do obey these laws. This assumption, however, is not justified at very low temperatures or very high pressures.

The study of materials in the liquid and gaseous states shows clearly that these states are in the nature of limiting conditions to gradual physical changes. This may be presented by means of Fig. 34. It should be clearly understood however that this figure is qualitatively but not quantitatively correct; that is, it is not drawn to scale, nor does it exactly represent the behavior of any real material. It does, however, show the *nature* of the changes under consideration for all known materials.

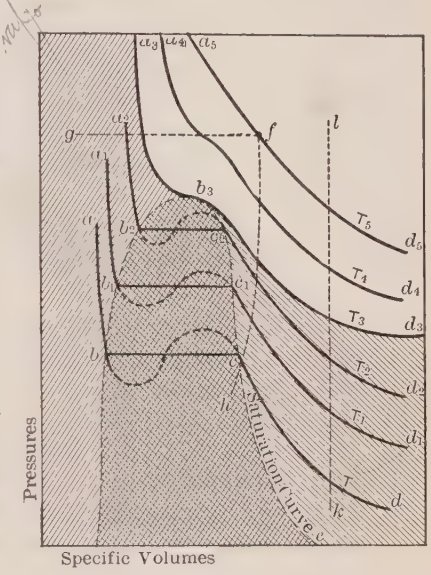


Fig. 34.— Isothermals of Material in Liquid, Vaporous and Gaseous States.

(b) The diagram is for unit weight of material on pressure-volume coördinates, and each of the heavy lines is an isothermal. Starting with the lowest line of the series, the point a represents the volume occupied by unit weight of liquid at temperature T and at the pressure shown. If the pressure is decreased while the temperature is maintained constant, the volume of the liquid will increase until the point b is reached.* At this point the pressure, volume, and temperature are such that any further change can only be a progressive vaporization at constant pressure, as shown by line bc (since the temperature is constant) with increase of volume from b to c; that is, the material is at the

* The increase of volume has been much magnified in the figure to emphasize the phenomenon.

point of vaporization for temperature T. At c the material has become fully vaporized, and hence is dry saturated vapor. A further decrease of pressure at constant temperature will cause it to become superheated and to behave somewhat like an ideal gas. The volume will then increase almost inversely with the pressure, bringing the material to the conditions d along the curve cd.

Starting from a_1, with the material in liquid form at a temperature $T_1 > T$, a similar process carries the material isothermally to d_1. The same statements can be made for all other starting points at different temperatures up to some such value as T_3, when the process will be that shown by the curve $a_3b_3d_3$. In this case the points b_3 and c_3 have become coincident, the liquid, if it is such, having the same volume at pressure P_{b3} as does its vapor.

At higher temperatures, such as T_4 and T_5, the material begins as a gas and the isothermals become more and more nearly rectangular hyperbolas ($PV = const$), as they are drawn for higher and higher temperatures.

Reversing the process, a gaseous material compressed isothermally from d_5 conditions will remain gaseous no matter how high the pressure is carried. A gaseous material compressed isothermally from d_1 will, however, begin to condense at c_1 and will continue to liquefy with further compression until it all becomes liquid at b_1.

(c) If d_3 is chosen as the point to begin isothermal compression, it is obvious that the material after passing b_3 must be on the boundary between the liquid and gaseous states; that is, the pressure, volume, and temperature conditions for the two states are the same and the material may be considered a liquid or a gas, or both.

The conditions at b_3 are called **critical conditions,** that is, critical volume, critical pressure, and critical temperature. The critical temperature of gaseous material is usually defined as the temperature above which liquefaction is impossible by any increase of pressure. The truth of this definition is evident from the diagram; no isothermal of higher temperature than $a_3b_3d_3$ could cross the latter and so enter the liquid region.

(d) In the figure the hatched area with the lines running upward from left to right represents the region in which the

material must be liquid. That is, when any point representing the pressure and volume of the substance falls within this region, the material must be in the liquid state. Similarly, the part hatched downward from left to right represents the region of superheated vapor, and that crosshatched in both directions represents the region of liquid mixed with its saturated vapor.

The part not hatched represents the region in which the material cannot be liquefied by change of pressure. This is now commonly called the region of the gaseous state. A gas may then be defined as a material above the critical temperature, and a vapor as material which, while resembling a gas, is below the critical temperature.

It must not be inferred that material above its critical temperature sensibly obeys the laws of ideal gases. It must be far removed on the temperature scale before this occurs. The isothermal T_5 shows this.

Note in the figure that the curve b_3e is the saturation curve, a part of which was drawn for water vapor in Fig. 32.

(e) This diagram, Fig. 34, is useful for determining the behavior of material when subjected to volume, pressure, and temperature changes. Material in the gas state, as at f for instance, can be liquefied by lowering temperature and decreasing volume while the pressure is maintained constant, as along the line fg. Or it can be brought to the condition of wet vapor by lowering pressure, volume, and temperature according to some curve fh. Similarly, increasing the temperature and pressure of a superheated vapor at constant volume (line kl) results in carrying it into the gas field.

(f) At the critical temperature the latent heat of vaporization, r, becomes zero; that is, no internal and no external work of measurable magnitude is done, as the material passes from just above to just below the point b_3 on the isothermal T_3. Inspection of the Steam Tables in the Appendix will show the way in which the latent heat of vaporization of water vapor gradually decreases from large values at low temperatures to a value of zero at the critical temperature.

76. Van der Waals' Equation for Real Gases. (a) Obviously any gas is really only a very attenuated liquid, differing in its properties from the liquid because its molecules are much farther

apart, and possibly of simpler structure. If this is true, it ought to be possible to write laws of condition which would fit the same material in either the liquid or the gaseous form. Several attempts have been made to do this, and one in particular is of great interest. It is due to Van der Waals and was developed by modifying Boyle's law to take account of two assumed facts. These are:

(1) The space filled by a gas is partly occupied by the molecules of that gas, and it is only the space between the molecules which obeys Boyle's law.

(2) In no real gas are the molecules far enough apart to be absolutely independent of one another; certain intermolecular forces still exist. These decrease the total volume occupied or make the gas behave as though subjected to a pressure greater than the real external pressure.

The law in mathematical form is

$$\left(P + \frac{a}{\mathbf{V}^2}\right)(\mathbf{V} - b) = \text{Constant},$$

or

$$\left(P + \frac{a}{\mathbf{V}^2}\right)(\mathbf{V} - b) = RT, \quad . \quad . \quad . \quad . \quad . \quad . \quad (117)$$

in which a and b are constants, differing with the kind of gas.

(b) This equation can be rearranged to read

$$\mathbf{V}^3 - \mathbf{V}^2\left(\frac{RT}{P} + b\right) + \mathbf{V}\frac{a}{P} - \frac{ab}{P} = 0, \quad . \quad . \quad . \quad (118)$$

a cubic equation in terms of the specific volume $\mathbf{V}$. Then for a given temperature and pressure there must be three values of $\mathbf{V}$ which satisfy the equation.

If the curves obtained by substituting in the equation are drawn for constant temperatures, they resemble the lines *abcd*, etc., in Fig. 34, except that the horizontal lines *bc*, etc., are replaced by the dotted curves shown. If the equation is really true, the process of vaporization must be more complicated than at first appears. The fact that the phenomena corresponding to part of the curve from *b* downward and from *c* upward can be realized experimentally gives evidence in support of this law. The condition of the material thus carried into the dotted position of the curve is, however, very unstable, and the sub-

stance suddenly assumes the condition shown by the horizontal line if disturbed.

(c) The critical point may now be said to be the point at which all three roots of the equation coincide or at which one is real and two are imaginary.

(d) The equation of Van der Waals, though better than that of Boyle, does not fully express the truth. If it did, it would hold for material in the solid as well as in the liquid state. It really recognizes no such condition as solid. If it did, its graph, continued far enough back in the direction *dcba*, would show another jog similar to but shorter than *cb*, representing the constant-pressure, constant-temperature change from liquid to solid. This it does not do, and hence it is imperfect.

(e) The phenomenon of zero volume at absolute zero temperature can now be explained. According to the simplest kinetic theory of gases, the temperature is supposed to be a measure of the translational energy of the molecules, and the pressure is the result of the bombardment of containing walls by the rapidly moving molecules.

Assuming, with Van der Waals, that the volume to which the ideal laws refer is not the total volume occupied by the gas, but equals that volume corrected for the volume of the molecules present, the limiting case of the ideal laws is easily explained. When absolute zero of temperature is reached the molecules of a gas must be assumed to be devoid of translational motion and in such positions that the volume referred to above has become zero. Then as the molecules at rest could not bombard surrounding surfaces the pressure would also be zero.

This equation of Van der Waals is in general only of theoretical interest to the engineer. Seldom does the accuracy required in engineering calculations warrant the use of such refinement. It is introduced here only to give a possible simple explanation, though an incomplete one, of what otherwise seems very indefinite, and to furnish a more complete view of the continuity of the liquid and gaseous states.

CHAPTER X.

PROPERTIES OF STEAM.

77. Steam or Water Vapor. (a) All that has just been said about the formation and the properties of vapors in general applies, of course, to the case of water vapor or steam.

This vapor is usually generated in a boiler in which the pressure is maintained substantially constant by the withdrawal of some of the steam as rapidly as more vapor is generated. This withdrawal occurs ordinarily through the steam pipe, at other times through the safety valve. The water when pumped into the boiler is under the pressure existing in that vessel. Thus the application of heat causes the temperature of the liquid to rise under constant-pressure conditions. This increase of course ceases when the temperature of vaporization, corresponding to the pressure, is reached. Since the heat is added at constant pressure, q would be computed from Eq. (94), using the mean specific heat at constant pressure for C_p.

(b) The further addition of heat to this water causes the formation of vapor, or steam. Associated with this process there is great increase in volume and the absorption of large amounts of heat. In discussing the general case, in connection with Fig. 30, it was considered that the *external latent heat* expended in connection with the volume-increase was utilized in lifting a weight, thus doing work in overcoming the action of gravity. In the case under consideration, when steam is supplied to a piston engine, the external latent heat is expended in displacing the piston against resistance, thus doing external work equal to APu per pound of material and making available increased volume of steam space in the cylinder as rapidly as the vapor is generated. It is true that engines of this type ordinarily take steam intermittently from the boiler, hence the steam pressure within that vessel would fluctuate slightly on this account, even if other causes of fluctuation could be eliminated. In such cases the mean pressure is the one commonly used.

It is not only true in the case of the piston engine but also in all other cases, that the withdrawal of steam from the boiler is accompanied by the doing of external work, equal to APu per pound of material, although just how this energy is expended is not always clear to the beginner.

The rest of the heat utilized in the process of vaporization is the "internal latent heat," ρ, expended in causing the molecular rearrangement accompanying the change from water to steam.

(c) In many instances a portion of the steam pipe is modified in form and subjected to heat in such manner that it becomes what is termed a "Superheater," in which the steam becomes superheated by the reception of more heat, as it passes through, on its way to the engine or other device which is being supplied. During this superheating the steam is under constant pressure, hence in using Eq. (102) to determine the heat added the mean specific heat at constant pressure $\bar{C}_p$ should be introduced.

78. Sources of Data. The different related properties of dry saturated steam are tabulated in Steam Tables such as that given in the Appendix. Some of the properties are determined directly by experiment and others are derived quantities which are found by computations involving the experimentally determined data.

Many different Steam Tables have appeared, and all except the most recent ones were based on Regnault's experiments, published in 1847. These older tables, while thus based on the same data, depart somewhat from one another in the values tabulated, the disagreement arising from differences in interpreting the data and in choosing values of Joule's equivalent, absolute zero, specific heat of liquid, etc.

In spite of their differences and errors, these steam tables are still sufficiently accurate for most engineering calculations; and ordinarily the results of investigation which involved their use may be compared with those based on the later tables, without introducing serious errors.

The recent rapid increase in the use of superheated steam has led to many attempts to determine accurately the different values of the specific heats of this material under various conditions. This has revived interest in the properties of saturated steam, with the result that in 1909 new and more accurate Steam

Tables appeared in book-form, one by Peabody, and another by Marks and Davis. Both books, besides giving tables for the properties of dry saturated steam, contain elaborate tables giving the entropy and other properties of superheated steam, and other auxiliary tables, together with certain charts which are useful to the engineer.

For the mechanical equivalent of 1 B.t.u., Peabody uses 778 foot-pounds and M. & D. use 777.52. For the absolute zero the former uses 491.5° F. below freezing; the latter 491.64. Peabody uses for the B.t.u. the heat required to raise one pound of water from 62° to 63° F.; whereas M. & D. use the "mean B.t.u." defined in Section 3.* The values chosen for the specific heats of water also vary slightly. However, the differences mentioned are so small as to be negligible for engineering purposes.

The discussion of how tables may be made will now be taken up very briefly. For a more thorough treatment and for references to the sources of data the student is referred to the books just mentioned.†

79. Properties of Dry Saturated Steam. The properties given in the Steam Table in the Appendix are the corresponding values of (**a**) Pressure and Temperature; (**b**) Heat of the Liquid; (**c**) Total Heat of Steam; (**d**) Latent Heat of Vaporization; (**e**) External Latent Heat; (**f**) Internal Latent Heat; (**g**) Entropies of Water, Vaporization, and Total; and (**h**) Specific Volumes.

The properties are tabulated for one pound of material, the pressures are in pounds per square inch absolute, and the heat quantities and entropies (excepting those for vaporization) are measured above 32° F.

Temperatures and Pressures.

(**a**) It has been seen that saturated vapor has a definite temperature corresponding to each pressure at which the vaporization occurs. The variation of temperature with pressure of water vapor has been determined experimentally and is shown graphically in Fig. 35, to two different pressure scales. It is important to note the shape of this curve, especially the rapid rise

* The mean B.t.u. is about $\frac{1}{900}$ larger than that measured at 62°.

† Also see Trans. A. S. M. E., Vols. 29 to 33, for papers, discussions, and references to sources.

of pressure, or increase in the slope $\frac{dp}{dt}$ with elevation of temperature in the upper region. The TP relations can also be

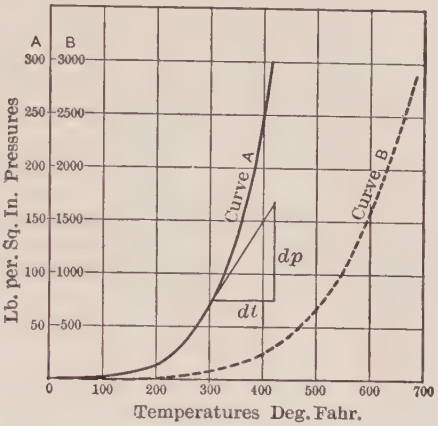


Fig. 35.—PT Relations for Steam.

expressed algebraically by formulas* which are rather complicated. These need not be given here, however.

Heat of Liquid (q).

(b) The heat of the liquid is the amount added to water at 32 degrees in order to bring it to the temperature of vaporization. Its amount is computed by using Eq. (92) and integrating between the temperatures of freezing and of vaporization, thus:

$$q = \int_{32}^{t_v} C_p\, dt = \int_{492}^{T_v} C_p\, dT, \quad . \quad . \quad . \quad . \quad (119)$$

where C_p is the constant-pressure specific heat of the liquid, which in the case of water varies with the temperature. The *progressive* values of C_p have been found by several experimenters with results that are not absolutely in accord. The curve in Fig. 36 represents an interpolation between the several data.

The right member of Eq. (119) will be recognized as the expression for the area below the C_p-curve, and lying between the ordinates at 32 degrees and t_v. This area can be found by planimeter or other method of integration.

* See "The Pressure-Temperature Relations of Saturated Steam," by Prof. Lionel S. Marks. Trans. A. S. M. E., Vol. 33.

If $\overline{C}_p$ is the **Mean Specific Heat** for the temperature range $d = (t_v - 32)$, between limits 32 degrees and t_v, then

$$q = \overline{C}_p\,(t - 32) = \overline{C}_p \times d. \quad . \quad . \quad . \quad . \quad (120)$$

$\overline{C}_p$ is obviously the mean height of the part of the C_p-curve lying between the temperature limits under consideration.

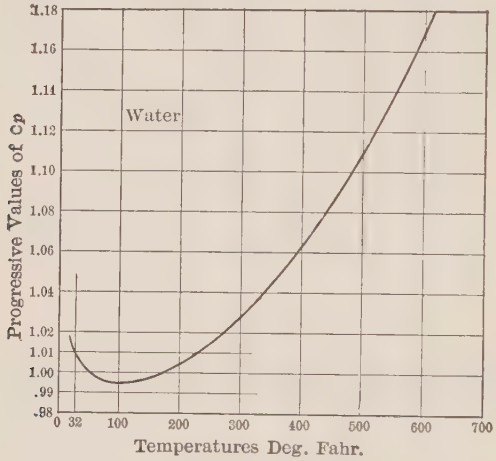


Fig. 36. — Progressive Values of Specific Heat, C_p, of Water.

Hereafter the instantaneous, or the progressive, values of C_p (that is, those corresponding to one degree rise at different temperatures) will be called the **progressive specific heats** to distinguish them from the mean values.

For many purposes, especially at low temperature, it is sufficiently accurate to assume $\overline{C}_p = 1$, then $q = (t - 32)$. In computing the values of q for the steam tables, however, it is necessary to employ the greatest accuracy.

In Fig. 31, the curve ab_3 shows approximately how q varies with t_v. If $\overline{C}_p$ is taken as unity, this curve becomes a straight line.

Total Heat of Steam (λ).

(c) This is the amount of heat required to raise one pound of water from 32 degrees to the temperature of vaporization, then to separate the constituent particles during the formation of

steam, and to do the external work accompanying the increase in volume.

The values of λ have been determined for a number of pressures by various experimenters. By plotting the most trustworthy data on cross-section paper, with λ and temperature as coördinates, Dr. H. N. Davis obtained a curve which is generally regarded as giving the most reliable values of this quantity. The portion of the curve lying between 212 degrees and 400 degrees is represented by the equation

$$\lambda = 1150.3 + 0.3745\,(t_v - 212) - 0.00055\,(t_v - 212)^2. \quad (121)$$

Regnault's formula for total heat, which was generally employed before 1909, is accurate enough for ordinary engineering purposes and is much simpler than Davis'. It is

$$\lambda = 1091.7 + 0.305\,(t_v - 32). \quad . \quad . \quad . \quad (122)$$

Note that this quantity increases with the temperature, but at a very slow rate. This is shown in Fig. 31, by the abscissas of points c, c_1, c_2, etc. The higher the pressure the less rapid is the rate of increase.

Latent Heat of Vaporization (r).

(d) Having obtained the total heat λ and the heat of the liquid q, the latent heat of vaporization may be found from

$$r = \lambda - q. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (123)$$

If the specific heat of water is taken as unity, $q = (t_v - 32)$; and if this is subtracted from Eq. (122), Regnault's approximate equation for the latent heat of vaporization is obtained. This is

$$r = 1091.7 - 0.695\,(t_v - 32). \quad . \quad . \quad . \quad . \quad (124)$$

In Fig. 31, the values of r for different temperatures and pressures are shown by the distances bc, b_2c_2, b_3c_3, etc. The latent heat decreases with rise in temperature, and becomes zero at the critical temperature. At atmospheric pressure r is 970,* and this figure should be remembered, as it is used frequently in engineering computations.

* The old value is 966.

The External Latent Heat (ΔE).

(e) The external latent heat ΔE expended in displacing the surrounding media can be computed from the equation

$$\Delta E = \frac{144\ pu}{778} = 144\ Apu = APu, \quad . \quad . \quad . \quad (125)$$

in which $A = \frac{1}{778}$, p is the pressure in pounds per square inch, P is the pressure in pounds per square foot, and u is the *increase* in volume during vaporization. How u may be determined will be explained in (h) of this section. The value of APu is relatively small and varies from about 61 B.t.u. at one pound pressure to about 85 B.t.u. at 400 pounds.

The Internal Latent Heat (ρ).

(f) The internal latent heat expended in producing the molecular rearrangement may be obtained by subtracting the external latent heat from the total. Thus

$$\rho = r - APu. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (126)$$

Entropies ($\Delta\phi$).

(g) The values tabulated are per pound of steam. The entropy of the liquid may be found from

$$\Delta\phi_l = \int_{492}^{T_v} \frac{dQ}{T} = \overline{C}_p \int_{492}^{T_v} \frac{dT}{T}. \quad . \quad . \quad . \quad . \quad (127)$$

As the heat of the liquid, $\overline{C}_p \int dT$, is measured above the freezing point of water, it follows that the corresponding entropy must also be calculated above the same datum, that is, 492° F. absolute. The integration of Eq. (127) gives for the **entropy of water,**

$$\Delta\phi_l = \overline{C}_p \log_e \frac{T_v}{492}, \quad . \quad . \quad . \quad . \quad . \quad (128)$$

in which T_v is the saturation temperature for the pressure under consideration and $\overline{C}_p$ is the *mean* specific heat of water for the temperature range from 32 degrees to t_v, as found from Fig. 36 in the manner described in Section 79 (b).

The **entropy of vaporization** ($\Delta\phi_v$) may be found from Eq.

(109) by substituting the values of T_v and r corresponding to the pressure under consideration.

The **total entropy** ($\Delta\phi_{sa}$) of one pound of dry saturated steam above 32 degrees is $\Delta\phi_{sa} = \Delta\phi_l + \Delta\phi_v$.

Specific Volume (V).

(h) This is the number of cubic feet occupied by one pound of steam. It varies with the pressure and is equal to the sum of the original volume of the pound of water (0.017 ± cu. ft.)* and u, the *increase* in volume during vaporization. Thus,

$$V = u + 0.017 \pm \text{cu. ft.} \quad . \quad . \quad . \quad . \quad (129)$$

The value of u can be obtained from what is known as Clapeyron's equation,

$$= \frac{r}{T_v} \cdot \frac{778}{\left(\frac{dP}{dt}\right)} \text{ cu. ft.} \quad . \quad . \quad . \quad . \quad . \quad (130)$$

Here $\left(\frac{dP}{dt}\right)$ is the slope of the pressure-temperature curve $\left(\text{see Fig. 35, in which } \frac{dp}{dt} = \frac{1}{144}\frac{dP}{dT}\right)$, and may be found either graphically or mathematically.

The following is a rather crude way of deriving Clapeyron's equation: On a PV-diagram, Fig. 37, starting at A with one

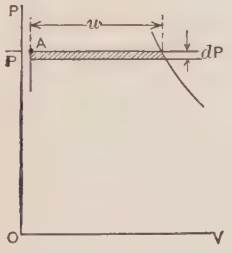


Fig. 37.

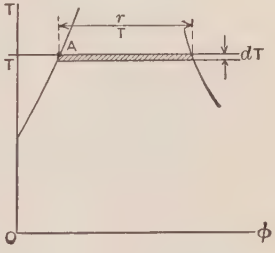


Fig. 38.

pound of water already at the boiling point (pressure P, and absolute temperature T), let sufficient heat be added to cause complete vaporization, the increase in volume being u; then let there be a slight drop in pressure dP, next let there be a de-

* The volume of a pound of water varies from 0.016 to 0.018 cubic feet within the ordinary range of temperatures.

crease in volume at the uniform pressure $(P - dP)$ until all of the steam is condensed to water at the corresponding boiling point; and finally bring the water up to its original temperature to complete the cycle. Evidently the work done, as shown by the area of the figure, is $u \cdot dP$ foot-pounds, which in B.t.u. is

$$dE = \frac{u \cdot dP}{778}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (a)$$

On the Tϕ-diagram, Fig. 38, let the same cycle be shown. Starting at A with water at the boiling temperature T, let heat, r, be added to vaporize the water. This is accompanied by an increase in entropy of amount $\frac{r}{T}$. Next let there be a temperature drop dT (corresponding to dP), and then let the steam be condensed at constant temperature $(T - dT)$, corresponding to $(P - dP)$, to water at the boiling point. Upon returning the water to its original condition the cycle is completed and the work done in B.t.u., as shown by the area surrounded, is

$$dE = \left(\frac{r}{T}\right) dT. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (b)$$

Evidently equations (a) and (b) both represent the same amount of work. Hence, $\frac{u\,dP}{778} = \left(\frac{r}{T}\right) dT$, solving which for u results in Clapeyron's equation.

The Specific Density.

(i) **The specific density** or weight of one cubic foot of steam is equal to $\left(\frac{1}{V}\right)$. As this is merely the reciprocal of the specific volume, it is not given in the Steam Table in the Appendix.

Properties of Steam at High Pressures.

(j) Above 250 pounds per square inch (400° F.) the properties of steam have not been determined with great accuracy, so that the values given in the tables above this pressure are not very trustworthy. More accurate values are, however, not available at present.

It will be noticed that the latent heat decreases as the temperature increases until it becomes zero at the **critical tempera-**

ture of about 706° F.,* corresponding to a pressure of about 3200 pounds per square inch.

80. Properties of Superheated Steam. (a) **Specific Heat at Constant Pressure.** In dealing with superheated steam the engineer ordinarily uses only the specific heat at constant pressure. For *ideal gases* it has been shown that C_p is independent of temperature and pressure, and that it is sensibly so for most real gases within ordinary ranges. For superheated steam, however, it cannot be considered constant at the temperatures used in engineering, for the material is always far below the critical conditions, and though approximating the behavior of a gas it varies greatly from the laws for perfect gases.

Several experimenters have recently determined values of C_p for steam over wide temperature and pressure ranges. Among these the results of Knoblauch and Jakob are generally considered the most trustworthy, and were used both by Peabody

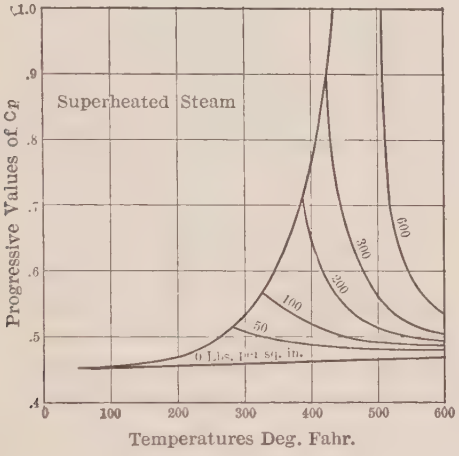


Fig. 39. — Progressive Values of Specific Heat C_p of Superheated Steam.

and by Marks and Davis in computing their tables. M. and D. made slight modifications to better coördinate the Knoblauch and Jakob results with those of other authoritative researches. The variation of the **progressive specific heat** C_p with temperature, for different constant pressures, is shown in Fig. 39. Be-

* Prof. L. S. Marks. Trans. A. S. M. E., Vol. 33.

cause of the comparatively rapid variation from degree to degree, the progressive values can be used in ordinary arithmetical calculations for a temperature rise of one degree only.

For greater ranges, the *mean specific heat* must be used, and this can be found from Fig. 39 in a manner similar to that described in **79** (**b**) for the mean specific heat of water. As most problems connected with superheated steam involve a temperature range D measured from the saturation temperature, T_v, for the pressure under consideration, it is convenient to have

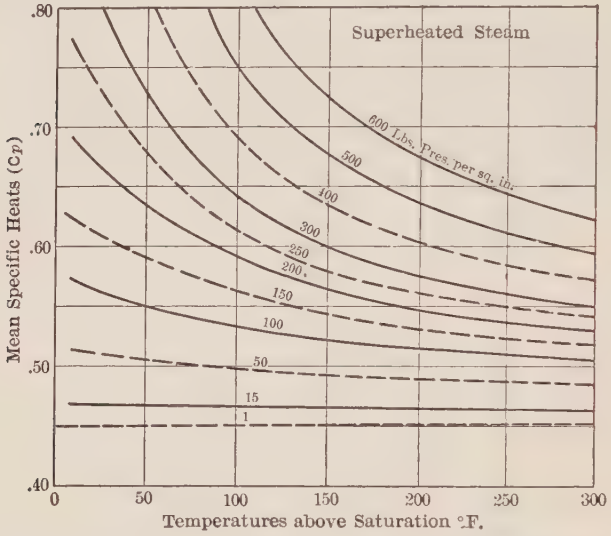


Fig. 40.—Variation of Mean Specific Heat $\overline{C}_p$ of Superheated Steam.

curves giving the constant-pressure **mean specific heat $\overline{C}_p$ measured above saturation temperature.** The values plotted in Fig. 40 may be used pending the appearance of more accurate ones.

Superheat.

(**b**) The heat added during superheating D degrees is evidently

$$\Delta Q_D = \overline{C}_p D, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (131)$$

where $$D = (T_{\text{sup}} - T_v).$$

The Total Heat of Superheated Steam (ΔQ_s).

(c) This quantity is the total heat above 32° F. per pound of steam which is superheated D degrees above saturation temperature. Representing this by ΔQ_s, it is given by the equation

$$\Delta Q_s = \lambda + \Delta Q_D = \lambda + \overline{C}_p D. \quad . \quad . \quad . \quad . \quad (132)$$

The Entropy of Superheated Steam.

(d) The **entropy above saturation** temperature T_v is $\Delta\phi_D$ and is given by Eq. (110).

The total entropy of steam superheated D degrees is obtained from Eq. (111) or (112).

Specific Volume of Superheated Steam (V_s).

(e) The volume of one pound of superheated steam may be computed from Linde's empirical formula

$$V_s = 0.5962\frac{T}{p} - (1 + 0.0014\,p) \times \left(\frac{150{,}300{,}000}{T^3} - 0.0833\right), \quad . \quad . \quad . \quad (133)$$

in which V_s is in cubic feet, T is the *absolute* temperature of the superheated steam in Fahr. degrees, and p is in pounds per square *inch*.

A simpler formula and one that is nearly as accurate is given by Tumlirz. It is, for p in *pounds* per square *inch*,

$$V_s = 0.5962\frac{T}{p} - 0.256, \quad . \quad . \quad . \quad . \quad . \quad (134)$$

and for P in pounds per square *foot*,

$$V_s = 85.86\frac{T}{P} - 0.256. \quad . \quad . \quad . \quad . \quad . \quad (135)$$

81. Temperature-Entropy Chart for Water and Steam. (a) Diagrams drawn with $T\phi$-coördinates are of great convenience in solving many problems involving the use of steam. Especially are they valuable when reversible adiabatic changes and associated heat changes are considered, for with these coordinates, the former are straight lines and the latter are areas.

The $T\phi$-chart may be constructed for any weight of working substance; but it is customary and more convenient to base it on

unit weight. The chart in Plate I in the Appendix is for one pound and the entropies are measured above 32° F. to correspond with the steam tables.

The value of a Tϕ-chart is greatly increased by the addition of certain lines of reference which aid in reading directly many of the quantities sought. The construction of these lines will now be considered.

Water Curve, or W-curve.

(b) Eq. (128) expresses algebraically the law by which the entropy of the liquid varies with the absolute temperature. From it can be obtained simultaneous values of T_v and $\Delta\phi_l$ and these can be used in plotting points on a $T\phi$-chart to show graphically the relation between the two variables. The Water Curve is the locus of these points and therefore is the graph of Eq. (128). In Fig. 41, AB is the W-curve.

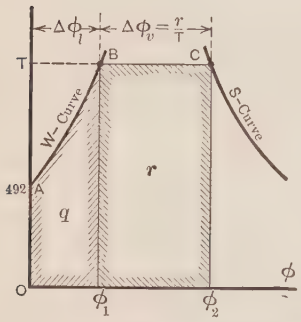


Fig. 41. — Tϕ-Diagram for Water Vapor.

If a steam table is available the values of $\Delta\phi_l$ and t_v used in plotting the W-curve can be obtained directly from it.

In general the heat used during a reversible process to produce a $T\phi$-change is

$$\Delta Q = \int_1^2 T\, d\phi.$$

The right side of this equation is of the form $\int y\, dx$, which is the mathematical expression for an area, and which here represents the heat quantities ΔQ. As dx in this case is $d\phi$, which is measured above 32° F., the *heat* represented by the area must be that *above* 32° F. also; and as $y = T$ (abs) these *areas must extend down to absolute zero of temperature*, that is, to the ϕ-axis. From this it is seen that the **heat of the liquid** above 32° F. is represented by the area under the W-curve, extending to the T and ϕ axes, such as area $OAB\phi_1$, in Fig. 41.

The W-curve has little curvature. If it is considered straight (involving small error for ordinary ranges), it is seen that the area under that line is the product of $\Delta\phi_l$ by the mean temperature $\frac{T+492}{2}$; that is,

$$q = \Delta\phi_l \cdot \frac{T+492}{2} \text{(approx.)} \quad . \quad . \quad . \quad . \quad . \quad (136)$$

Substituting $q = (T - 492)$, which would be its value when $C = 1$, gives the following approximate equation for the entropy of water:

$$\Delta\phi_l = \frac{2\,(T-492)}{T+492} \text{ (approx.)}, \quad . \quad . \quad . \quad (137)$$

which is convenient for rough computations, as it does not involve the use of tables.

Saturation Curve, or S-curve.

(c) The entropy of dry saturated steam is, from Eq. (115), $\Delta\phi_{sa} = \Delta\phi_l + \Delta\phi_v = \Delta\phi_l + r/T$, the values of all quantities in which are given in the steam tables. In Fig. 41, the abscissa TB is $\Delta\phi_l$ for the temperature T; so if BC is made equal to the corresponding value of $\Delta\phi_v$, the point C must fall on the Saturation Curve. The locus of a series of points plotted in this manner for different temperatures is the S-curve. Evidently this curve is the graph of Eq. (115).

The area of the rectangle below the line BC is

$$\Delta\phi_v \times T = \frac{r}{T} \times T = r,$$

and hence is the **latent heat of vaporization.** Then the **total heat of the steam,** λ, is given by the area below ABC, since this latter represents $r + q$.

Constant-Quality Curves, or X-curves.

(d) The equation of these curves is $\Delta\phi_x = \Delta\phi_l + \frac{xr}{T}$, in which x is constant for each curve and is equal to the quality under consideration. Taking various corresponding values of r and T from the steam tables, the quantities $\left(\frac{xr}{T}\right)$ may be computed, and add-

ing these to the values of $\Delta\phi_l$ for the corresponding temperatures gives $\Delta\phi_x$. In Fig. 42, TB as before equals $\Delta\phi_l$ and BD is

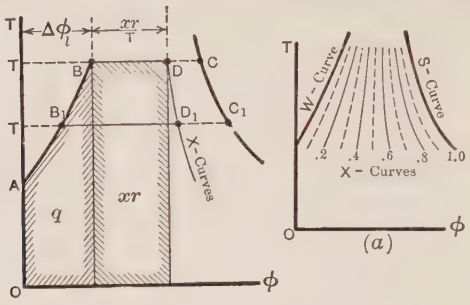


Fig. 42.—Tϕ-Diagram Showing X-Curves.

laid off equal to $\left(\frac{xr}{T}\right)$, thus locating the state point at D for the temperature T. The locus of points similarly plotted for different temperatures is the Curve of Constant Quality. A series of such curves is shown in Fig. 42 (*a*).

Since $BC = \frac{r}{T}$ and $BD = \frac{xr}{T}$, it follows that $\frac{BD}{BC} = x$. This relation suggests another and simpler method of plotting points to determine the X-curve: In the figure draw the horizontal intercepts BC, B_1C_1, etc., between the W-curve and the S-curve, and on them locate the points D, D_1, etc., in such positions that $\frac{BD}{BC} = \frac{B_1D_1}{B_1C_1} = etc. = x$. Then the locus of these points, D, D_1, *etc.*, is the curve desired.

The heat used in vaporizing x parts of a pound of steam at temperature T is shown by the area below BD, Fig. 42, since this area $= \left(\frac{xr}{T}\right) \times T = xr$. The total heat in the mixture of steam and water is given by the area below ABD, for this area equals $xr + q$.

Constant-Volume Curves, or V-curves.

(e) At any temperature T, Fig. 43, the change in entropy from B to C during complete vaporization is accompanied by an increase, equal to u, in the volume of the working substance. If at the same temperature only part of the unit weight — occupying the volume V — is in the vaporous form, it is evident

that the quality of the steam must be $x = \frac{(V - 0.017)}{u}$. By maintaining V constant in this equation and substituting values of u corresponding to different temperatures, the way x varies with T during an isovolumic change can be determined. Then the V-curve can be plotted either by using the quality or by making

$$\frac{BD}{BC} = \frac{(V - 0.017)}{u}, \qquad \frac{B_1D_1}{B_1C_1} = \frac{(V - 0.017)}{u_1}, \text{ etc.}$$

The same curve can be obtained by graphical construction in the following manner: In Fig. 43 lay off a V-axis opposite to the T-axis, thus forming a Vϕ-quadrant in which volumes are laid off downward. Directly below B drop an ordinate ab for the corresponding volume of the material. This is the volume of the water at temperature T, or 0.017 ± cubic feet. Directly below C lay off the volume corresponding to that point, thus locating c. The value in this case is **V**, the specific volume of the steam. Then the straight line bc joining these points shows the uniform increase of the volume and entropy during the process of vaporization of one pound of working substance at the temperature T. In like manner similar Vϕ-lines, such as b_1c_1, b_2c_2, etc., can be drawn for other temperatures of vaporization. If any isovolumic line V' is then drawn, it intersects bc, b_1c_1, etc., at points v, v_1, v_2, etc., whence projecting upward to the corresponding isothermals determines points D, D_1, etc., on the V-curve with $T\phi$ coördinates. In the case shown in this figure, $V' = \mathbf{V}_1$, so v_1 would coincide with c_1, and D_1 with C_1.

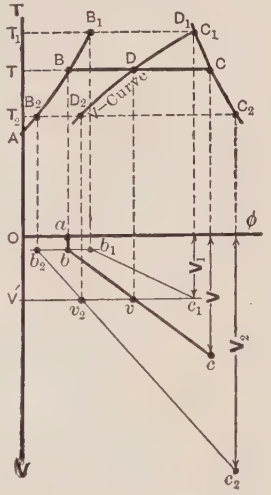


Fig. 43.—Tϕ-Diagram Showing Method of Constructing V-Curves.

Constant-Heat Curves, or Q-curves.

(f) For **wet saturated steam** the equation of this curve is $xr + q = \text{const.} = \Delta Q$. For any given ΔQ, the variation of x with T can be found by substituting the values of r and q corre-

sponding to the different temperatures used. Several of these curves for different values of ΔQ are shown in Fig. 44. Referring to curve $E\,E_1E_2$, etc., it is evident that the areas under ABE, AB_1E_1, etc., must all be the same and equal to the value of ΔQ for that curve. Note particularly that these curves represent $xr + q$ and not $x\rho + q$.

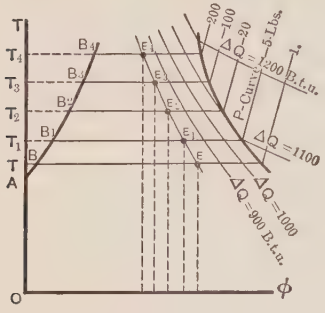


Fig. 44. — Tϕ-Diagram Showing P-Curves, and Q-Curves.

(g) For **superheated steam** the Q-curve is found in the following manner: For any assumed pressure p, the corresponding values of λ and T_v are obtained from the Steam Tables. Then if ΔQ is the constant-heat quantity under consideration, the temperature rise during superheating at pressure p is $D = \dfrac{\Delta Q - \lambda}{\overline{C}_p}$. The ordinates of the Q-curve are $T = T_v + D$, and the abscissas are

$$\Delta\phi_s = \Delta\phi_{sa} + \overline{C}_p \log_e \frac{T_v + D}{T_v}, \quad . \quad . \quad . \quad (138)$$

in which T_v, D and $\overline{C}_p$ are known and $\Delta\phi_{sa}$ can be obtained from the tables.

A difficulty arises in selecting the proper value of $\overline{C}_p$, because the mean specific heat is dependent on D, which is initially unknown. Hence it is necessary to adopt the "cut and try method." That is, a trial value of $\overline{C}_p$ is assumed and D is computed; then the value of $\overline{C}_p$ corresponding to the pressure and to D is obtained from the curves, and if it is the same as the trial value the assumption was correct; but if there is much difference, a new value must be assumed and the process must be repeated.

Constant-Pressure Curves, or P-curves.

(h) For **saturated steam** the P-curves are isothermals; for **superheated steam** they are not. In the latter case, the relation between $\Delta\phi_s$ and $(T_v + D)$, the temperature after superheating, is given by the Eq. (138). If this is solved for any

fixed pressure, T_v and $\Delta\phi_{sa}$ become constants, and the variables are $\Delta\phi_s$ and $(T_v + D)$ with related values of $\overline{C}_p$. Corresponding values of these variables would be used in plotting the P-curves, several of which are shown in Fig. 44.

The temperature of saturation T_v for any pressure can be found by using these curves, for it is given by the ordinate of the point of intersection between the corresponding P-curve and the Saturation Curve.

The Final Tϕ-chart.

(i) The final Tϕ-chart, Plate I in the Appendix,* contains all the curves described in this section, and to it has been added a scale for the absolute pressures corresponding to the temperatures of saturation.

For a point anywhere on it in the Saturation Region there can be read directly the corresponding values of T_v, $\Delta\phi_x$, x, V, p, and ΔQ. The latter is given either by the Q-curve or by area; the values of q and xr are given by areas; and the pressures can be read either on the scale at the left or by extending the isothermal to intersect the S-curve, thus finding the corresponding P-curve in the superheated region.

If the point is on the W-curve, T_v, $\Delta\phi_l$, and p can be read directly, while q is given by the area below the curve.

For a point on the saturation line the values of T_v, $\Delta\phi_{sa}$, λ, $\mathbf{V}$, and p can be read at once.

If the point is in the Region of Superheat, T, $\Delta\phi_s$, $\Delta\phi_{sa}$, p, and ΔQ can be read directly; the increase in temperature above saturation is $D = T - T_v$; the B.t.u. superheat, ΔQ_D, is given either by an area, or by $(\Delta Q - \lambda)$; and the entropy of superheat $\Delta\phi_D$ is $(\Delta\phi_s - \Delta\phi_{sa})$.

If any expansion line is drawn on the chart, all of the above-mentioned quantities can be read for each point on the line. If the lines inclosing a cycle are drawn, the work done per cycle is of course given by the area surrounded.

It is important to note that the quantities given by the Q-curves are values of $(xr + q)$, not $(x\rho + q)$, and contain the external work of vaporization $(xAPu)$.

* A larger and more accurate Tϕ-chart is contained in Peabody's Steam and Entropy Tables, published by Wiley & Sons.

82. The Mollier Chart, or Qϕ-Chart. (a) This chart, Fig. 45, is constructed with Qϕ-coördinates. On it are drawn lines for constant pressure (P-curves); for constant qualities (X-curves)

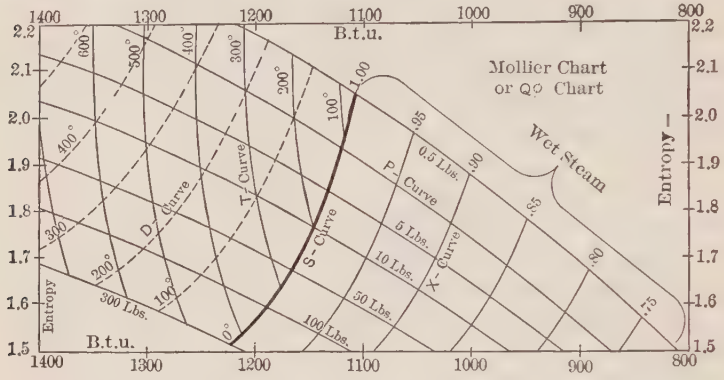


Fig. 45.—Mollier or Heat-Entropy Chart.

for wet steam; and for constant temperatures (T-curves) for superheated steam. The boundary line between the Regions of Saturation and Superheat is the Saturation Line (S-curve).

(b) For wet steam $\Delta Q_x = xr + q$ and $\Delta\phi_x = \Delta\phi_l + x\Delta\phi_v$. If the pressure is constant, x, ΔQ_x, and $\Delta\phi_x$ are the only variables. Then by substituting different qualities, related values of ΔQ_x and $\Delta\phi_x$ may be found and these can be used in plotting the P-curve. A series of such curves is shown in the figure.

(c) Lines joining points of the same quality on the different P-curves constitute the X-curves.

(d) For superheated steam

$$\Delta Q_s = \lambda + \overline{C}_p D \text{ and } \Delta\phi_s = \Delta\phi_{sa} + \overline{C}_p \log_e \frac{T_v + D}{T_v}.$$

If the pressure is constant, ΔQ_s, $\Delta\phi_s$, and $(T_v + D)$ are the variables. By substituting different values of D in these equations, the ΔQ_s and $\Delta\phi_s$ coördinates of points on the P-curve may be found.

(e) Lines through points of like temperatures are the T-curves, and as drawn these give the temperatures in degrees Fahr., not absolute. Lines through points representing the same increase of temperatures above saturation constitute D-curves.

(f) The final Qϕ-chart is given in Plate II in the Appendix. This has all the curves just discussed, except the D-curves.

For any point on it there can be read at once the values of ΔQ, $\Delta\phi$, p, and x (or t).

82A. The Ellenwood QV-Chart. (a) In this chart, Fig. 45a, the coördinates are total heat (ΔQ) and volumes ($x\mathbf{V}$ or $\mathbf{V}_s$) per pound. Oblique lines are given respectively for constant values of pressure (p), quality (x), superheat (D) and entropy (ϕ).

(b) For any state point on the chart one may read directly the values of p, x or D, $x\mathbf{V}$ or $\mathbf{V}_s$, $\Delta\phi$ and ΔQ. From the intersection of the pressure line with the q-curve the corresponding value of q may be read and from the temperature scale along this curve the vaporization temperature t_v may be determined. Hence in addition to the values given by the Mollier Chart, this one gives volumes per pound, q, and t_v, which greatly increases its field of application.

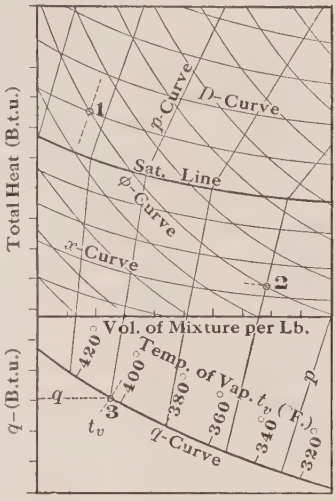


Fig. 45a.—Ellenwood Chart.

(c) Plate IV in the Appendix is a small two-page Ellenwood Chart * in which the volume scale changes progressively, which accounts for the scolloped appearance of the curves.

82B. The Constant-Pressure External-Work Chart (Ellenwood). (a) In Plate III (Appendix)* the abscissas are volumes and the ordinates give the external work of formation of steam, *under constant pressure*, from one pound of water at 32° F. For any point on the chart one may read the external work, the volume per pound, the pressure, and the quality (or superheat).

(b) The Tϕ, Mollier, and Ellenwood Charts give *total* heat. To obtain the **intrinsic heat** necessitates subtracting the external work (computed or obtained from Plate III).

* Redrawn to greatly reduce scale from "Steam Charts" by F. O. Ellenwood, published by John Wiley & Sons, Inc.

CHAPTER XI.

VOLUME CHANGES OF VAPORS.

83. General. Saturated and superheated vapors, like gases, may be made to change volume in many different ways, but the general study of such transformations may be based on a few simple cases. The laws governing these changes are different from those for similar gas processes, and this is because of the different properties of the materials dealt with. For convenience the order of treatment in this chapter is different from that of Chapter V.

84. Constant-Pressure and Isothermal Volume-Changes for Saturated Vapors. (a) Fixing the pressure of a saturated vapor, wet or dry, fixes the temperature; hence a constant-pressure change of such material must also be a constant-temperature, or isothermal one.

The line *ab* in the PV-diagram, Fig. 46, and in the Tϕ-diagram, Fig. 47, is the graph of an isobaric or isothermal change for dry saturated vapor. In Fig. 46, the abscissa of point *a* represents the volume of unit weight of the liquid at the temperature of vaporization corresponding to the pressure P_1. By the addition of heat the liquid can be vaporized to any desired extent until finally it has all become dry saturated vapor. This is an expansion at constant pressure and at constant temperature and is the only isothermal expansion possible with saturated vapor. It follows that isobaric and isothermal volume changes of saturated vapors can only occur during vaporization or condensation of the material. This is equivalent to saying that such transformations are always accompanied by quality changes.

Such volume changes cannot be carried beyond a quality of 100 per cent, because then the material will all be saturated steam, with the specific volume corresponding to the existing pressure, and because further addition of heat at constant pressure must increase the volume of unit weight above the value at saturation, hence must superheat the vapor.

Equation for Isobaric and Isothermal Changes of Saturated Vapors.

(b) The equation of such changes in terms of P and V must be the same as that for the constant-pressure change of gases; that is,

$$PV^0 = P = \text{Constant.}$$

There is, however, a real difference in the two cases. When dealing with gases it is possible, in imagination at least, to carry the isobaric expansion to any desired volume, while in the case

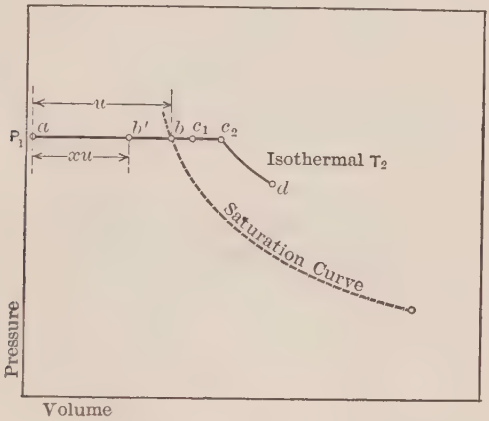


Fig. 46. — PV-Diagram for Vapor.

of saturated vapors expansion per pound cannot be carried beyond the specific volume corresponding to the existing pressure without changing the nature of the material and its behavior.

The volume occupied by *one pound* of material depends on the quality x, and can be computed for *water and its vapor* from

$$V' = 0.017 + xu = x\mathbf{V} \text{ (approx.)}.$$

This is true, no matter what process the material has undergone, and a similar equation can be found for each material.

Heat Changes during Isobaric or Isothermal Changes of Saturated Vapors.

(c) If the expansion starts with all the material as liquid at the temperature of vaporization, that is, with an initial vapor volume equal to zero, the heat change is merely that accompany-

ing vaporization, and must equal the latent heat of vaporization per pound of material if the condition of dry saturation is reached. Hence the heat added is

$$\Delta Q = r = (\rho + APu), \quad . \quad . \quad . \quad . \quad . \quad (139)$$

where u is the volume change represented by the distance ab in Fig. 46 and ΔQ is the area below ab in Fig. 47. In the case of water vapor, the values of all the quantities occurring in this equation may be obtained from the Steam Table given in the Appendix.*

If the pound of material is not completely vaporized but has a quality equal to x, then

$$\Delta Q = xr = x\rho + xAPu, \quad . \quad . \quad . \quad . \quad (140)$$

in which xu is the volume change, which is shown by the distance ab' in Fig. 46 and ΔQ is the area below ab_1 in Fig. 47.

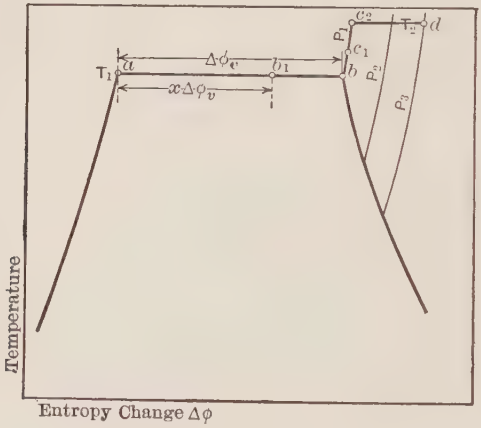


Fig. 47.—Tϕ-Diagram for Vapor.

If the expansion is from quality x_1 to x_2, with corresponding volume change from x_1u to x_2u (not shown in the figure), the case is general, and the change in associated heat is

$$\Delta Q = x_2r - x_1r$$
$$= (x_2 - x_1)\ (\rho + APu) \quad . \quad . \quad . \quad . \quad . \quad (141)$$

* For steam, $u = (V - 0.017\ \pm)$, in which V may be obtained from the Steam Tables.

Work during Isobaric or Isothermal Changes of Saturated Vapors.

(d) The External Latent Heat of vaporization is that part of the total heat which does the external work accompanying the increase of volume; it must therefore be equivalent to the external work done. Hence in vaporizing to quality x, per pound of material,

$$\Delta E = x \cdot APu \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (142)$$

and

$$778\,\Delta E = x \cdot Pu \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (143)$$

This work is shown in Fig. 46 by the area below ab'. For the case in which $x = 1.00$, it is the area below ab.

With change of quality from x_1 to x_2 the work done is,

$$\Delta E = (x_2 - x_1)\, APu \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad (144)$$

and

$$778\,\Delta E = (x_2 - x_1)\, Pu \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (145)$$

85. Constant-Pressure Volume Changes of Superheated Vapors. (a) Starting from the point b in Figs. 46 and 47, the dry and saturated vapor may be made to still further expand at constant pressure to some point c_1 by superheating, that is, by raising the temperature above that corresponding to saturation at that pressure. The further this expansion continues the more nearly the behavior resembles that of a gas, and there is no theoretical limit to such expansion, as there was in the case of the saturated vapor.

Equation of Isobaric Change of Superheated Vapor.

(b) The equation in terms of P and V must be the same as that already developed for gases and saturated vapors, namely,

$$PV^0 = P = \text{Constant.}$$

Heat Change during Isobaric Changes of Superheated Vapor.

(c) As the temperature must be raised at constant pressure in order to increase the volume, or lowered at constant pressure to reduce the volume, it follows that a quantity of heat equal to the specific heat at constant pressure, C_p, must be added or abstracted per degree change. Then for heat added *above saturation*

$$\Delta Q_D = \int_1^2 C_p\, dT,$$

or, using the mean specific heat $\overline{C}_p$ for the temperature range D measured from the temperature of saturation,

$$\Delta Q_D = \overline{C}_p D. \quad . \quad . \quad . \quad . \quad . \quad . \quad (146)$$

In Fig. 47 this is shown by the area below bc_1 for a case in which the heat change is reversible.

For an isobaric change from superheat temperature D_1 to D_2

$$\Delta Q = (\overline{C}_p D)_2 - (\overline{C}_p D)_1. \quad . \quad . \quad . \quad . \quad . \quad (147)$$

This is equivalent to $\Delta Q = \overline{C}'_p (D_2 - D_1) = \overline{C}'_p (T_2 - T_1)$, where $\overline{C}'_p$ is the mean specific heat for the temperature range involved. In Fig. 47 this heat change is shown for reversible conditions by the area below c_1c_2.

The foregoing equations giving the heat change are not sufficient for use in engineering problems as they generally occur. It is usually not only necessary to know the range of superheat, but also the volume change accompanying it. In the case of gases, this can be found from the Law of Charles, but superheated vapors as generally treated in engineering are not far enough removed from the condition of saturation to even sensibly obey that law.

It is possible to find this volume change for **superheated water vapor** by using the approximate equation of Tumlirz previously given as Eq. (134). Writing this for volumes $\mathbf{V}_1$ and $\mathbf{V}_2$ and then dividing gives, for pressures in pounds per square *inch*,

$$\frac{\mathbf{V}_1}{\mathbf{V}_2} = \frac{T_1 - 0.4293\, p_1}{T_2 - 0.4293\, p_2}. \quad . \quad . \quad . \quad . \quad . \quad (148)$$

Since $p_2 = p_1$ during an isobaric change, this equation reduces to the form

$$\frac{\mathbf{V}_1}{\mathbf{V}_2} = \frac{T_1 - \text{Constant}}{T_2 - \text{Constant}}, \quad . \quad . \quad . \quad . \quad . \quad (149)$$

in which the constant has a different value for every pressure.

The effect of this corrective constant becomes less with increase of temperature or decrease of pressure, and there is an accompanying closer approach of the equation to that of Charles' Law and a closer resemblance of the superheated vapor to an ideal gas.

Work during Isobaric Changes of Superheated Vapor.

(d) During an isobaric change with one pound of any working material the work is

$$778\,\Delta E = P\,(\mathbf{V}_2 - \mathbf{V}_1) \text{ ft.-lbs.}, \quad . \quad . \quad . \quad (150)$$

which was first developed in the case of gases as Eq. (24). If, in Fig. 46, the expansion is from c_1 to c_2, the work done is given by the area below c_1c_2.

86. Isothermal-Volume Changes of Superheated Vapors. (a) These must in a general way resemble the isothermal changes of gases, since superheated vapors approximate the gaseous conditions. The exact behavior of any particular vapor under these conditions must, however, be determined by experiment. For superheated water vapor the necessary information can be obtained from the approximate equation of Tumlirz.

Equation of Isothermal Change of Superheated Water Vapor.

(b) Rearranging the Tumlirz equation (134) and maintaining T constant gives the following for isothermal changes for this material, for p in pounds per square *inch*,

$$p\mathbf{V} + 0.256\,p = 0.5962\,T = \text{Constant}, \quad . \quad . \quad (151)$$

and from Eq. (135), for P in pounds per square *foot*,

$$P\mathbf{V} + 0.256\,P = 85.86\,T = \text{Constant}. \quad . \quad . \quad (152)$$

Comparing this latter with the Eq. (14) for gases, namely,

$$P\mathbf{V} = RT,$$

it is evident that it differs only in the addition of a second term ($0.256\,P$) in the first member. Obviously the smaller the numerical value of the pressure the smaller will be this corrective factor and the more nearly will the vapor obey the ideal gas law. If Eqs. (151) and (152) be divided by T, it becomes apparent that the greater the numerical value of T the smaller will be the effect of the corrective factor and therefore the more nearly will the material approach the condition of an ideal gas.

Work during Isothermal Changes of Superheated Vapor.

(c) The work done by an expanding vapor, as well as gas, is given in all cases by the expression first developed as Eq. (41),

$$778\,\Delta E = \int_{V_1}^{V_2} P\,dV \text{ ft.-lbs.}$$

However, in order to perform the integration in any case, it is necessary to know the relation between P and V, and this is a matter for experimental determination. The relation given by the equation of Tumlirz, Eq. (135), may be used for superheated water vapor. From this equation,

$$P = \frac{85.86\ T}{\mathbf{V} + 0.256},$$

which value may be substituted in the type integral, and the integration performed, giving per *pound* of water vapor

$$778\ \Delta E = 85.86\ T \int_{V_1}^{V_2} \frac{d\mathbf{V}}{\mathbf{V} + 0.256}$$

$$= 85.86\ T \log_e \frac{\mathbf{V}_2 + 0.256}{\mathbf{V}_1 + 0.256} \text{ ft.-lbs.} \quad (153)$$

Eq. (43b), for work during isothermal changes of gases, may be written

$$778\ \Delta E = \left(P_1 \mathbf{V}_1 \log_e \frac{\mathbf{V}_2}{\mathbf{V}_1}\right) = RT \log_e \frac{\mathbf{V}_2}{\mathbf{V}_1}.$$

Comparing Eq. (153) with this, it again appears that the higher the temperature and the lower the pressure the more nearly do the equations developed for superheated water vapor approach those for the behavior of gases, and by analogy the same must be true of the behavior of all superheated vapors.

If, in Fig. 46, the isothermal expansion is from c_2 to d, the work is represented by the area below $c_2 d$.

Heat Change during Isothermal Changes of Superheated Vapors.

(d) Applying Eq. (1),

$$\Delta Q = \Delta S + \Delta I + \Delta E,$$

to this case, it is evident that the term ΔS must be zero, since temperature does not change; but ΔI must have some value other than zero, since the materials cannot be said to even sensibly approach the condition of ideal gases. The term ΔE must also have a value other than zero if work is to be done by or upon the superheated vapor. For any case ΔE can be readily found, but ΔI is more difficult to evaluate, and any equation for the value of ΔQ which could be developed would necessarily be a very cumbersome one.

(e) Fortunately the Tϕ-diagram offers a simple means of determining ΔQ, since this quantity is represented by an area on that diagram, when the change is reversible.

Assume for instance that it is desired to find the heat required when one pound of water vapor is expanded isothermally and reversibly from a pressure P_1 and a temperature T_2 above saturation temperature T_{v_1}, to a lower pressure P_3. It is only necessary to draw the horizontal line c_2d in Fig. 47, between the two pressure lines and at the desired temperature, and then determine the area under c_2d.

Fig. 47 is only a special case of Plate I of the Appendix, and in practice the latter would be used.

(f) Note that the heat added to, or subtracted from, superheated steam isothermally is not equal to the difference in total heats, since the isothermal of superheated vapors is not a constant-pressure line. In the case of reversible isothermal expansion this heat is equal to $T \times \Delta\phi$, both of which quantities can be obtained from the Mollier or Ellenwood Charts given in the Appendix.

87. Adiabatic Changes of Saturated Vapors. (a) With the exception of problems involving the flow of vapors in which the material is accelerated as a whole, the adiabatic changes of vapors which are considered by the engineer are thermodynamically reversible in the ideal case. In the following paragraphs only these *reversible* processes will be considered, leaving the more complicated irreversible processes for later development.

Since reversible adiabatic changes are also isentropic ones, their graph on the Tϕ-diagram must be a vertical line. This offers a very easy means of studying these changes in every case where there are sufficient experimental data for the drawing of this diagram.

(b) The diagram in Fig. 48 is developed from the Tϕ-diagram for *water vapor* with the lines of constant quality shown — originally given in Fig. 42 (*a*) and Plate I. To this have been added vertical lines representing *reversible adiabatic expansions* starting at each 20 per cent of quality at the pressure P_1. The diagram shows that *when the initial quality is high* (point *a* in Fig. 48) *the quality of water vapor must decrease as the expansion progresses, and when the initial quality is low* (point *b*) *it must in-*

crease during expansion. Near the middle of the diagram, that is, with initial quality near 50 per cent, x remains nearly con-

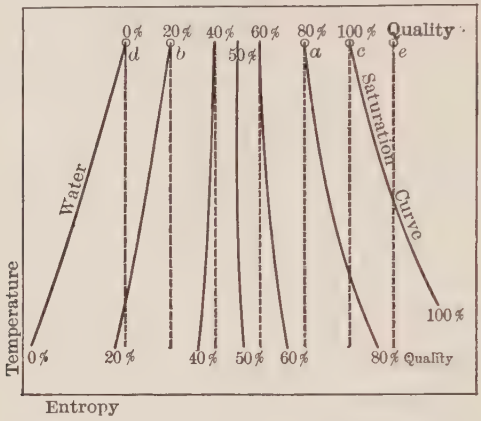


Fig: 48. — Tϕ-Diagram for Water Vapor.

stant during the entire expansion. This is not necessarily a property of all vapors, as it depends on the relation between the various heat quantities and is thus a matter for experimental determination.

This is well shown by considering the case of *Ether Vapor.* The Tϕ-diagram for this material is given in Fig. 49. As in the

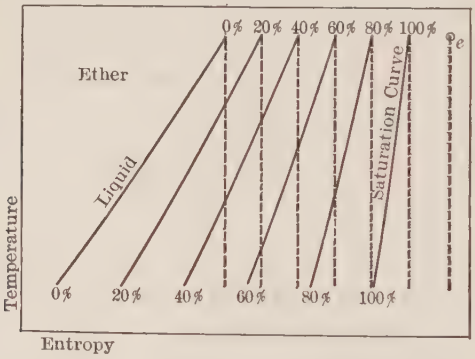


Fig. 49. — Tϕ-Diagram for Ether Vapor.

last case, the constant-quality lines and isentropic lines for every 20 per cent initial quality are drawn. It is evident from the figure that during reversible adiabatic expansion of ether vapor

the quality must continually increase, whatever its initial value may be.

(c) At any initial quality between 0 per cent and 100 per cent the quality changes will be governed by the relative quantities of liquid and vapor present, as can be seen for the case of water by referring to Fig. 48. In this case large quantities of liquid make available so much heat with decrease of pressure or temperature that evaporation (or "quality increase") must occur as expansion progresses; large quantities of vapor make available so small a quantity of heat by pressure drop alone that condensation must occur.

With initial qualities of about 50 per cent, the two effects approximately balance and the quality remains almost constant.

(d) The case of ether as already shown is very different. Referring to Fig. 49, it is seen that with expansion starting at any quality between 0 per cent and 100 per cent the heat liberated due to pressure drop alone is more than sufficient to do the necessary external work, and the expanding material must absorb part of the liberated heat. When the quality is less than 100 per cent, this is done by vaporization, with possible superheating toward the end of the process; and when the quality is equal to or greater than 100 per cent, superheating occurs throughout the expansion.

Comparison of Figs. 48 and 49 shows that because the heat of the liquid varies much more rapidly and is much greater in quantity in the case of ether vapor, the quality lines for that vapor all slope in the same direction, thus accounting for the difference in phenomena occurring during adiabatic expansion.

Equation of Reversible Adiabatic Changes of Saturated Vapors.

(e) Since these changes are generally studied by means of the $T\phi$-diagram, the most useful equation is

$$\Delta\phi = \text{Constant, or } (\Delta\phi_x)_1 = (\Delta\phi_x)_2.$$

This equation gives no direct means of plotting the curves representing adiabatic expansion to PV-coördinates, but may be used indirectly for that purpose.

First, the quality at the end of adiabatic expansion, from pressure p_1 and quality x_1 to pressure p_2, may be computed by solving for x_2 in the following equation:

$$(\Delta\phi_l + x\,\Delta\phi_v)_1 = (\Delta\phi_l + x\,\Delta\phi_v)_2. \quad . \quad . \quad . \quad (154)$$

Then, the volume occupied by *unit* weight of the substance, at the end of the expansion, is found by multiplying the specific volume by x_2.

As already shown, the isentropic line corresponding to Eq. (154) gives a means of reading directly the quality of the expanding material on the Tϕ, Mollier, and Ellenwood Charts.

(f) For water vapor at common operating pressures and with initial quality between 100 and 70 per cent, the relations between pressure and volume during adiabatic expansion are given approximately, but very accurately, by the equation

$$PV^n = \text{Constant}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (155)$$

in which the value of n is given by the following equation,

$$n = 1.035 + 0.1\,x, \quad . \quad . \quad . \quad . \quad . \quad . \quad (156)$$

where x is the initial quality expressed as a decimal fraction.

The PV relations can also be obtained from the Ellenwood and Tϕ Charts.

Work Done during Adiabatic Changes of Saturated Vapors.

(g) Since all the work done during such an expansion must be obtained at the expense of intrinsic heat energy, and since no heat energy is used for other purposes, it follows that if $(xr + q - xAPu)_1 = (x\rho + q)_1$ represent the intrinsic heat energy before an adiabatic change and $(xr + q - xAPu)_2 = (x\rho + q)_2$, the intrinsic heat energy after such a change, the External Work Done is

$$\Delta E = (xr + q - xAPu)_1 - (xr + q - xAPu)_2 \; . \quad (157)$$

$$= (x_1\rho_1 + q_1) - (x_2\rho_2 + q_2). \quad . \quad . \quad . \quad . \quad (158)$$

In using this equation the initial conditions are known: x_2 is obtained from Eq. 154, and ρ_2 and q_2 are found from the Vapor Tables for pressure p_2.

(h) If the **PV-diagram,** Fig. 50, be for one pound of steam, then when the point b is reached the heat-energy $(xAPu)_1$ has been abstracted from the steam and absorbed by displacing a piston or surrounding media against resistance. Thus there remain $(x\rho + q)_1$ heat units with which to begin the adiabatic expansion. At point c, there are $(x\rho + q)_2$ heat units left in the steam, and the quantity $(xAPu)_2$ would not appear unless either by compression or some equivalent process the volume of the steam is contracted isobarically an amount x_2u_2 to the volume of the liquid, as shown at d.

The area below ab is $(xAPu)_1$ B.t.u.; the area below bc shows the work done, or heat utilized during adiabatic expansion alone,

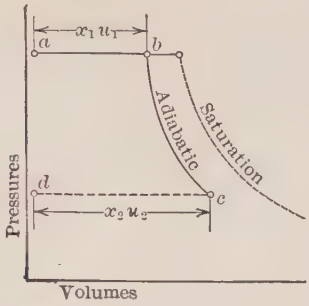


Fig. 50. — PV-Diagram.

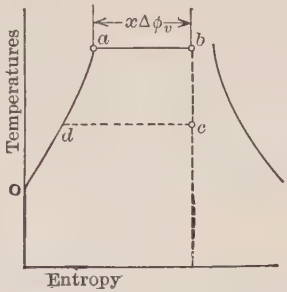


Fig. 51. — Tϕ-Diagram.

and is $[(xp + q)_1 - (xp + q)_2]$ B.t.u.; the area below cd is $(xAPu)_2$ B.t.u.

(i) On the **Tϕ-diagram,** Fig. 51, the areas below lines such as oab represent $(xr + q)$ quantities which include the external latent heat of vaporization. From these quantities must be deducted the appropriate values of $xAPu$* to obtain the heat in the steam during an isentropic process. (Note that in the Tϕ-chart, Plate I, the $xAPu$ quantities are also included in the values given by the Q-curves.)

(j) On the **Mollier diagram** and on the **Ellenwood Chart** the abscissas (ΔQ) also include the external work, and this latter * must be deducted when considering the heat utilized *during* adiabatic expansion alone (see (g) above).

88. Adiabatic Changes of Superheated Vapors. (a) These changes, like those for saturated vapors, are best studied by means of the Tϕ-diagram. Vertical lines, such as that through e in Fig. 48, drawn to represent reversible adiabatic expansion of superheated water vapor, show that as the expansion is carried to lower pressures the material approaches the saturated condition and may indeed attain a quality less than unity. On the other hand, similar lines on the Tϕ-diagram for ether, Fig. 49, show that if such an expansion starts with superheated vapor the superheat increases as the expansion continues.

* The constant-pressure external work of formation of steam from one pound of water can be obtained from the External Work Chart, Plate III, in the Appendix.

Equation of Reversible Adiabatic Changes of Superheated Vapors.

(b) As in the case of saturated vapors, the general equation for reversible adiabatic changes of superheated vapor is

$$\Delta\phi = \text{Constant, or } (\Delta\phi_s)_1 = (\Delta\phi)_2.$$

If the steam is expanded to wetness, the quality may be found by solving for x_2 in the following equation:

$$(\Delta\phi_s)_1 = (\Delta\phi_l + x\Delta\phi_v)_2. \quad . \quad . \quad . \quad . \quad (159)$$

$(\Delta\phi_s)_1$ can be computed from Eq. 138, $(\Delta\phi_l)_2$ and $(\Delta\phi_v)_2$, in the case of water vapor, can be obtained directly from the Steam Tables.

If the expansion takes place entirely in the Region of Superheat, the final temperature $T_2 = (T_v + D)_2$ can be found from

$$(\Delta\phi_s)_1 = \left(\Delta\phi_{sa} + \overline{C}_p \log_e \frac{T_v + D}{T_v}\right)_2. \quad . \quad . \quad (160)$$

Here $\overline{C}_p$ is the mean specific heat for the temperature range $D_2 = (T - T_v)_2$ and T_{v_2} is the temperature of vaporization at the terminal pressure.

External Work Done during Adiabatic Changes of Superheated Vapors.

(c) As in other cases of adiabatic changes, the external work done during this reversible adiabatic change is equal to the intrinsic heat change. During a constant-pressure change from liquid at the temperature of vaporization to superheated vapor, the external work per pound is

$$\Delta E_p = AP(\mathbf{V}_s - 0.017) \text{ B.t.u.}, \quad . \quad . \quad . \quad (161)$$

where $\mathbf{V}_s$ is the specific volume of the superheated steam from Eq. (134); hence the external work done, if the steam remains in the superheated state throughout the isentropic expansion, is

$$\Delta E_s = \left[\lambda + \int_{T_v}^{T} C_p\, dT - \Delta E_p\right]_1 - \left[\lambda + \int_{T_v}^{T} C_p\, dT - \Delta E_p\right]_2 \quad (162)$$

$$= [\lambda + \overline{C}_p D - \Delta E_p]_1 - [\lambda + \overline{C}_p D - \Delta E_r]_2, \quad . \quad . \quad . \quad . \quad (163)$$

in which D_2 is found from Eq. (160)

If vapor initially superheated is expanded to wetness with quality x_2, the external work done is

$$\Delta E_{sx} = [\lambda + \overline{C}_p D - \Delta E_p]_1 - [x\rho + q]_2, \quad . \quad . \quad . \quad (164)$$

in which x_2 is found from Eq. (159).

On the PV-diagram this work is represented by the area below the expansion line. In using either the Tϕ, Mollier, or Ellenwood diagrams, to obtain the work done *during* isentropic expansion alone, it is necessary to deduct the ΔE_p quantities (and the APu quantities if entering the saturation region) from the heat values found.

89. Constant-Volume Changes of Saturated Vapors. (a) If a saturated vapor is to change pressure at constant volume, there must be a quality change, because the same weight of material in the form of vapor cannot occupy a given space at two different temperatures. During a pressure drop there is a tendency for saturated vapor to increase in volume, hence if the volume is maintained constant there must be a decrease of quality; that is, condensation must take place. The reverse is of course true for a pressure rise.

Equation of Constant-Volume Change of Saturated Vapor.

(b) As in previous cases, the equation of a constant-volume change is, in terms of pressure and volume,

$$V = \text{Constant}.$$

Heat Change during Constant-Volume Change of Saturated Vapor.

(c) It was shown in Section 70 that the quality of saturated vapor could be found by dividing the volume occupied per pound of mixed vapor and liquid by the specific volume corresponding to the pressure existing. By using this method the quality changes which occur during a constant-volume change of saturated vapor may be found, and when the quality at any pressure is known the intrinsic heat for that state may be determined.

90. Constant-Volume Changes of Superheated Vapors. (a) When a superheated vapor changes pressure at a constant volume, there must be a temperature change similar, but not equal, to that occurring in the case of an ideal gas undergoing

the same sort of change. The equation of Tumlirz, Eq. (134), may be used to find the temperature of water vapor corresponding to any pressure and volume, and hence such changes (or their equivalents, if irreversible) can be plotted to PV or $T\phi$ coördinates.

Equation of Constant-Volume Changes of Superheated Vapors.

(b) As in all other cases, the equation in terms of PV coördinates is

$$V = \text{Constant.}$$

Heat Change during Constant-Volume Change of Superheated Vapor.

(c) Since the temperature and pressure can be found for any point in a constant-volume pressure change of superheated vapor, the intrinsic heat can also be found for every point. The difference between the intrinsic heats at beginning and end of the constant volume change must be the amount of heat added to, or subtracted from, the steam.

CHAPTER XII.

VAPOR CYCLES.

91. Carnot Cycle with Dry Saturated Steam, (a) The Carnot cycle may be carried out with a saturated vapor of any kind in the same apparatus as was assumed in Section 49 and shown in Fig. 17. For simplicity assume the cylinder to contain unit weight of water at the temperature T_1. Then the volume occupied by the liquid, inclosed by the cylinder head, cylinder walls, and piston, will be that of unit weight of water at temperature T_1 and corresponding pressure. This is plotted as the point a on the PV-diagram, in Fig. 52, with volume greatly exaggerated.

If heat is added to the liquid from the hot body U and the piston is allowed to move out at just the proper rate to maintain a constant pressure on the working substance, vaporization will occur at constant pressure and therefore at constant temperature. The volume would consequently increase isothermally, or the process would be an isothermal expansion.

When vaporization is complete the volume attained will be the specific volume, V_b, of water vapor at temperature T_1 and corresponding pressure. The isothermal expansion will then be represented by the constant pressure line ab.

If now the nonconducting cylinder cover Z is applied and the piston allowed to continue its outward motion, the expansion of the vapor must be adiabatic. The actual shape of the line representing such expansion will be given approximately by $PV^n = \text{Constant}$, and is represented by the curve bc, on which c is a point where the temperature has reached that of the cold body, T_2.

If now Z is replaced by the cold body and the piston is forced inward, condensation must occur, the heat liberated being absorbed by the cold body. Condensation, like evaporation, is a change at constant temperature and constant pressure, and hence is represented by a horizontal line from c toward the left.

To complete a Carnot cycle, it is necessary to stop the process of condensation when the volume has decreased to some value V_d, so chosen that the final adiabatic compression will bring the material back to the liquid form with conditions T_1, P_a, and V_a.

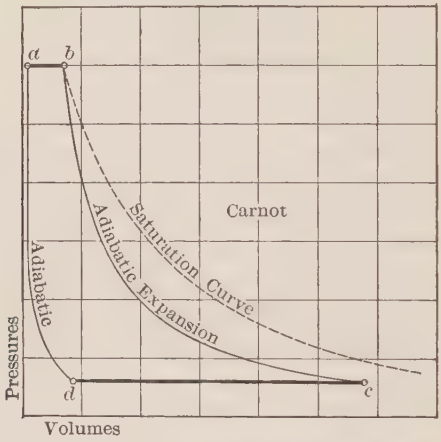


Fig. 52. — PV-Diagram for Carnot Cycle with Dry Saturated Water Vapor at *b*.

(**b**) The Tϕ-diagram of the cycle is drawn in Fig. 53, on which the water curve and saturation curve are indicated by dotted lines. This diagram is lettered to correspond with Fig. 52. It shows how the quality of the steam must decrease during the adiabatic expansion *bc*, and how by stopping the condensation, or isothermal compression, at the proper point, *d*, it is possible to return the material to the liquid condition at temperature T_1 by adiabatic compression *da*.

Note that the Tϕ-diagrams for the Carnot cycle for vapor and gas are identical, but that this is not true for the PV-diagrams because of the difference in the properties of the materials.

Work per Pound of Dry Saturated Water Vapor Operating in Carnot Cycle.

(**c**) The work done per cycle can be obtained in several ways, two of which will be considered. They are practically the same as those previously used for gas cycles.

(**d**) The *first method* is to take the algebraic sum of the quantities of work done during the several processes of the cycle.

(1) The work during the isothermal expansion equals $\frac{P_1}{778}(V_b - V_a)$ B.t.u., and (2) that during the isothermal compression similarly equals $\frac{P_2}{778}(V_c - V_d)$ B.t.u. (3) The work during adiabatic expansion must be, as shown in Eq. (158), the difference between the quantities of intrinsic heat energy above 32° F. at the beginning and end of the process; that is, $(q_b + \rho_b) - (q_c + x_c\rho_c)$. (4) Similarly, the work during adiabatic compression is $q_a - (q_d + x_d\rho_d)$. The values of the qualities x_c and x_d can be found from the constant-entropy equation (154) or from either

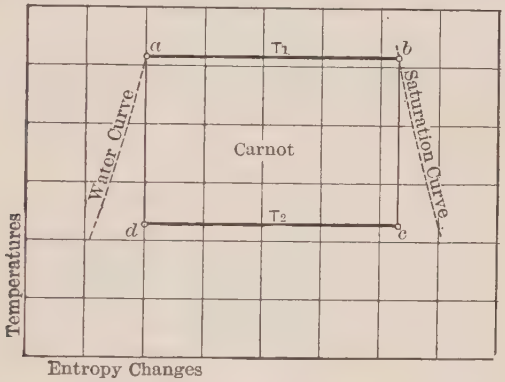


Fig. 53. — Tϕ-Diagram for Carnot Cycle with Dry Saturated Water Vapor at *b*.

of the entropy diagrams; hence in any problem all the terms are known and the total work done during the cycle equals the algebraic sum of the four expressions.

(e) The *second method* and more direct one is to subtract from the total heat supplied the total heat rejected; the difference must be the heat converted into work, and must be represented by the area within the four lines of the cycle.

The heat supplied during the isothermal expansion is r_1, the latent heat of vaporization of the material at the temperature T_1. The heat rejected is the part of the latent heat liberated during the partial condensation and is $(x_c r_2 - x_d r_2) = r_2(x_c - x_d)$, in which x_c and x_d are determined from the constant-entropy equation* (154) or from either of the entropy diagrams. The

* Just as x_c is the quality at the end of adiabatic expansion *bc*, so x_d may be considered as the quality at the end of an adiabatic expansion *ad*. The constant-entropy equation is applied to the line *bc* to find x_c and to the line *ad* to find x_d.

external work done must then be, when the **steam is dry and saturated** at the beginning of expansion,

$$\Delta E = \Delta Q_1 - \Delta Q_2 = r_1 - r_2 (x_c - x_d) \text{ B.t.u.} \quad . \quad . \quad (165)$$

and

$$778 \Delta E = 778 [r_1 - r_2 (x_c - x_d)] \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (166)$$

The expression numbered (165) is really obvious from the Tϕ-diagram drawn in Fig. 53.

(f) From Fig. 53 it is also seen that

$$\Delta E = (T_1 - T_2) \Delta \phi_v = (T_1 - T_2) \frac{r_1}{T_1} \quad . \quad . \quad (167)$$

The last form is the simpler in use. In it the expression $\frac{(T_1 - T_2)}{T_1}$ is the efficiency Ef_c of the cycle, as will be shown next, hence

$$\Delta E = r_1 \times Ef_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (168)$$

Efficiency of the Carnot Cycle Using Dry Saturated Water Vapor as a Working Substance.

(g) The efficiency must of course equal the ratio of the work done per cycle to the heat supplied per cycle; hence from Eq. (165)

$$Ef_c = \frac{r_1 - r_2 (x_c - x_d)}{r_1} .$$

A more convenient expression can be found directly from the Tϕ-diagram.

(h) Remembering that area under the line *ab* in Fig. 53 represents heat supplied from the hot body, and that the area of the cycle represents heat converted into work, it is evident that

$$Ef_c = \frac{T_1 (\Delta \phi_b - \Delta \phi_a) - T_2 (\Delta \phi_c - \Delta \phi_d)}{T_1 (\Delta \phi_b - \Delta \phi_a)} = \frac{T_1 - T_2}{T_1} , \quad . \quad (169)$$

which is the same as the expression for efficiency of the Carnot engine using gas as a working substance.

(i) The Carnot cycle, consisting as it does of two reversible isothermals crossed by two reversible adiabatics, must have identical Tϕ-diagrams for all working substances. Since the development just given depends only on this diagram and not

upon the properties of the material, it follows that the expression $\frac{T_1 - T_2}{T_1}$ must give the efficiency of the Carnot cycle operating with any working substance.

92. The Carnot Cycle with Any Vapor. (a) The case just considered, in which the working vapor is brought to the dry saturated condition before adiabatic expansion begins, is the simplest possible case as far as the expressions for heat and work are concerned. But adiabatic expansion might begin with the liquid only partly vaporized by isothermal expansion; that is, with a quality, x_b, at the top of the adiabatic. Or, the vapor might be superheated before adiabatic expansion begins. Further, a material like ether, with properties markedly different from steam, and with different behavior during adiabatic expansion, might be used.

In any case

$$Ef_c = \frac{T_1 - T_2}{T_1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (170)$$

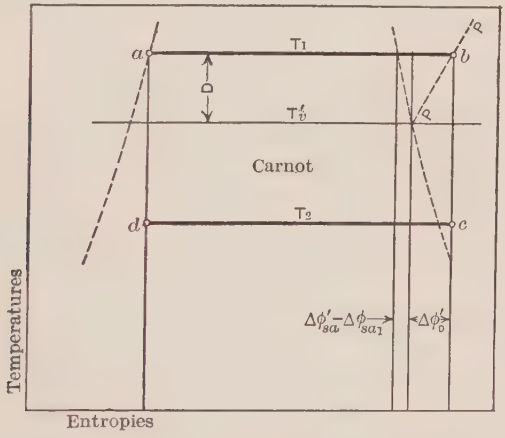
and

$$\Delta E = \Delta Q_1 \times Ef_c, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (171)$$

where ΔQ_1 is the heat added at constant temperature T_1 to the liquid previously raised to that temperature.

For steam initially **dry and saturated,** $\Delta Q_1 = r_1$ . . . (172)

For steam initially **wet,** $\Delta Q_1 = x_1 r_1$ . . (173)



[Fig. 54.—Tϕ-Diagram for Carnot Cycle with Superheated Steam.

(b) For steam at pressure P, Fig. 54, **superheated** to temperature T_1, ΔQ_1 is given by the area below ab. In the figure PP

is the constant-pressure curve through b and $\Delta\phi_{sa}'$ is the entropy of saturation at this pressure. Evidently

$$\Delta Q_1 = r_1 + T_1(\Delta\phi_{sa}' - \Delta\phi_{sa_1} + \Delta\phi_D'). \quad . \quad . \quad . \quad . \quad (174)$$

$$= r_1 + T_1\left(\Delta\phi_{sa}' - \Delta\phi_{sa_1} + \bar{C}_p' \log_e \frac{T_v' + D}{T_v'}\right). \quad (175)$$

where the subscript 1 refers to the values corresponding to T_1 and the primed quantities are those referring to the pressure at the point b: hence r_1 does not correspond to pressure P.

(c) In Fig. 55 are shown PV- and Tϕ-diagrams of Carnot cycles illustrating different possibilities when saturated and

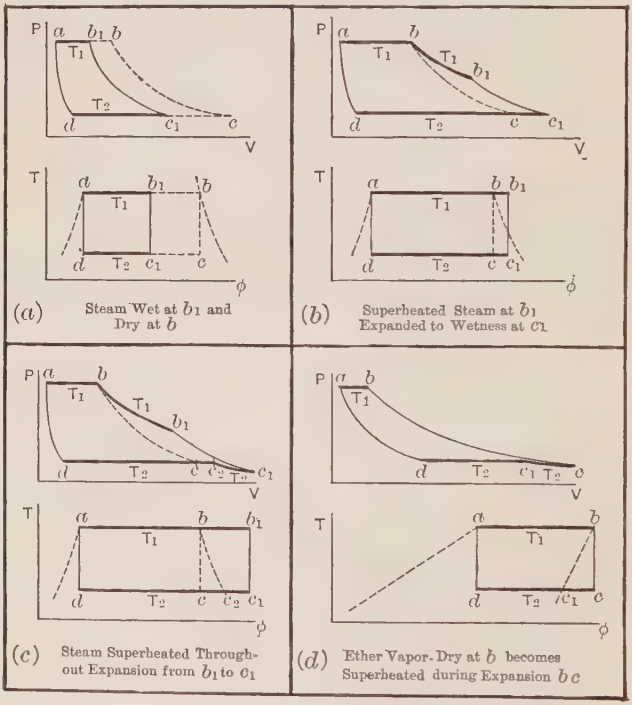

(a) Steam Wet at b_1 and Dry at b
(b) Superheated Steam at b_1 Expanded to Wetness at c_1
(c) Steam Superheated Throughout Expansion from b_1 to c_1
(d) Ether Vapor-Dry at b becomes Superheated during Expansion bc

Fig. 55. — Carnot Cycles for Vapors — Various Possibilities.

superheated vapor are used as working substances. The bold lines represent the isothermal reception and rejection of heat. In the PV-diagram these are horizontal only when the vapor is saturated. Figs. 55 (*a*), (*b*), and (*c*) are for steam, and in each

case *abcd* is the diagram which is obtained with dry saturated vapor at *b*. Fig. 55 (*d*) is for ether.

93. Clausius Cycle with Dry Saturated Water Vapor. (a) This cycle is often called the Rankine cycle, but as another cycle which is universally known by this latter name must also be considered, the name of Clausius will be used in this book to designate the cycle at present under consideration. As shown in the PV-diagram, Fig. 56, it consists of two constant-pressure

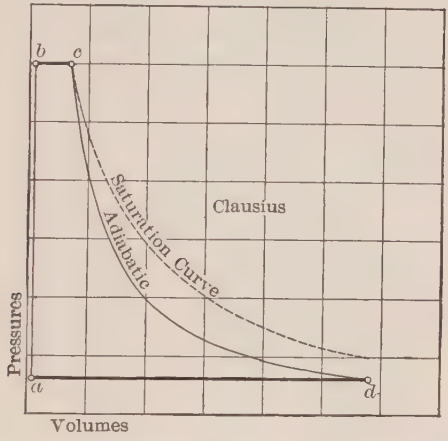


Fig. 56. — PV-Diagram for Clausius Cycle with Dry Saturated Water Vapor at *c*.

lines *bc* and *da* joined by an adiabatic *cd* and what is practically a constant-volume line *ab*. The apparatus of Fig. 17 used in developing the Carnot cycle can also be used for the ideal Clausius cycle.

The volume plotted at *b* is that of unit weight of water just ready to vaporize, corresponding to *a* of the Carnot cycle shown in Fig. 52. The addition of the latent heat of vaporization, r_1, causes the material to expand at constant pressure until it occupies the specific volume $\mathbf{V}_c$ at *c*. This quantity of heat, as before, comes from the hot body at temperature T_1.

The adiabatic expansion is exactly like that of the Carnot cycle and is produced in exactly the same way.

The constant-pressure decrease of volume starts exactly like the similar line in the other cycle, but condensation is carried to completion by the removal of heat equal to $x_d r_2$. The volume

V_a is then the volume of unit weight of water at the temperature of vaporization corresponding to the lower pressure P_a. The heat given up during this condensation is received by the cold body at the constant temperature T_2.

The line *ab* which takes the place of the adiabatic compression of Carnot represents the heating of the liquid from temperature T_2 to the higher value T_1, while the pressure rises from P_a to P_b. There will be a very small volume change in the liquid during this process, but it is so small in comparison with the other volume changes in the cycle that it may be neglected and the process considered as a constant-volume pressure rise.

(b) The Tϕ-diagram correspondingly lettered is shown in Fig. 57. The heat used to raise the temperature of the water must

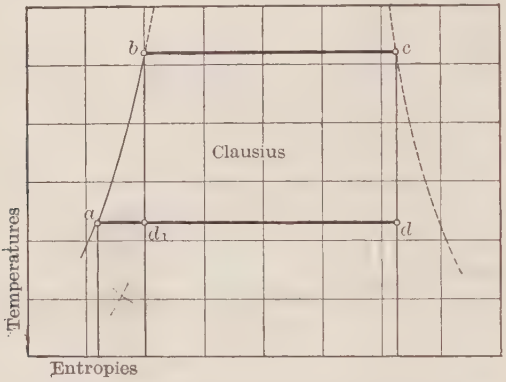


Fig. 57. — Tϕ-Diagram for Clausius Cycle with Dry Saturated Water Vapor at *c*.

come from the hot body which has the temperature T_1, and during its reception the temperature of the water will vary from T_2 to T_1. Hence the cycle does not fulfill the criterion for maximum efficiency because *all heat is not received when the working substance is at its highest temperature*. It is also evident that the cycle is not reversible, because the addition of heat to the liquid exemplifies a process which is intrinsically irreversible. Strictly interpreted, the line *ab* in Fig. 57 represents a reversible process equivalent to the irreversible process *ab* of the Clausius cycle.

Work per Pound of Water Vapor Carried through Clausius Cycle with Dry Saturated Vapor at Beginning of Adiabatic Expansion.

(c) As before, $\Delta E = \Delta Q_1 - \Delta Q_2$, from which the value of the work done per cycle may be determined. The heat ΔQ_1 consists of two parts, (1) that added to raise the temperature of the water from T_2 to T_1, and (2) the heat used in vaporizing during the volume change from V_b to V_c. The quantity ΔQ_2 given up during the condensation, as already explained, can be determined as soon as the quality x_d is known. This is easily found from Eq. (154) or from either of the entropy diagrams.

Then,

$$\Delta E = \Delta Q_1 - \Delta Q_2 = \{(q_b - q_a) + r_1\} - \{x_d r_d\} \text{ B.t.u.} \quad . \quad (176)$$

$$= \lambda_1 - q_2 - x_d r_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (177)$$

From inspection of the Tϕ-diagram it is evident that the work done is given by the following expression, the symbol $\overline{C}_p$ standing for the mean specific heat of the liquid over the temperature range:

$$\Delta E = \overline{C}_p (T_1 - T_2) + T_1 (\Delta\phi_c - \Delta\phi_b) - T_2 (\Delta\phi_d - \Delta\phi_a). \quad . \quad . \quad . \quad . \quad (178)$$

Since $(\Delta\phi_c - \Delta\phi_b) = \Delta\phi_{v_1}$, $\Delta\phi_d = \Delta\phi_{sa_1}$, and $\Delta\phi_a = \Delta\phi_{l_2}$, Eq.(178) may be written

$$\Delta E = \{\overline{C}_p (T_1 - T_2) + T_1 \Delta\phi_{s_1}\} - \{T_2 (\Delta\phi_{sa_1} - \Delta\phi_{l_2})\}. \quad . \quad . \quad . \quad . \quad . \quad (179)$$

A more useful formula, which may also be written from inspection of the Tϕ-diagram, is

$$\Delta E = \frac{r_1}{T_1}(T_1 - T_2) + q_1 - q_2 - T_2 (\Delta\phi_{l_1} - \Delta\phi_{l_2}), \quad . \quad (180)$$

all quantities in which may be obtained directly from the Steam Tables.

Efficiency of the Clausius Cycle with Dry Saturated Water Vapor at the Point c.

(d) To find this item, it is only necessary to divide the work done, Eq. (177), by the heat supplied; then

$$Ef. = \frac{\Delta E}{\Delta Q_1} = \frac{\lambda_1 - q_2 - x_d r_2}{\lambda_1 - q_2} \quad . \quad . \quad . \quad . \quad (181)$$

$$= 1 - \frac{x_d r_2}{\lambda_1 - q_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (182)$$

(e) This form is not readily comparable with the expression for the Carnot efficiency, and although the fact is already known that the Clausius efficiency must be lower than the other because of the addition of heat below maximum temperature, it is of interest to derive an expression which will show this difference. This can be done by using Eq. (178) in obtaining the efficiency expression, thus,

$$Ef. = \frac{\Delta E}{\Delta Q_1} = \frac{\bar{C}_p (T_1 - T_2) + T_1 (\Delta\phi_c - \Delta\phi_b) - T_2 (\Delta_{\psi d} - \Delta\phi_a)}{\bar{C}_p (T_1 - T_2) + T_1 (\Delta\phi_c - \Delta\phi_b)}$$

$$= 1 - \frac{T_2 (\Delta\phi_d - \Delta\phi_a)}{\bar{C}_p (T_1 - T_2) + T_1 (\Delta\phi_c - \Delta\phi_b)} \quad . \quad . \quad . \quad (183)$$

$$= 1 - \frac{\Delta Q_2}{\Delta Q_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (184)$$

The Carnot efficiency written in similar form is

$$Ef. = 1 - \frac{T_2}{T_1} = 1 - \frac{\Delta Q_2}{\Delta Q_1} \quad . \quad . \quad . \quad . \quad . \quad (185)$$

In Eqs. (184) and (185), the magnitude of the last term determines the value of the efficiency in each case, but inspection of the expressions as they stand does not show which of the last terms is the greater. If Fig. 57, which shows the two cycles superposed, is consulted, the interpretation of the last terms is much simplified.*

It is evident from the figure that the heat supplied during the Clausius cycle, equal to the area under *abc*, is greater than that

* In the strict interpretation of Fig. 57, the line *ab* is not the irreversible line of the Clausius cycle, but represents a reversible process which would give the same P, V, T conditions as the other, as mentioned before in connection with Fig. 57.

supplied during the Carnot cycle by the triangular area abd_1, plus the area below ad_1. The heat rejected is, however, greater by the area below ad_1. Therefore in the case of the Clausius cycle the heat rejected is increased in greater proportion than the heat received, and the fraction $\frac{\Delta Q_2}{\Delta Q_1}$ for this cycle must be greater than for the Carnot, and hence the efficiency is less.

94. The Clausius Cycle in General. (a) As in the case of the Carnot cycle, it is possible to imagine a Clausius cycle developed

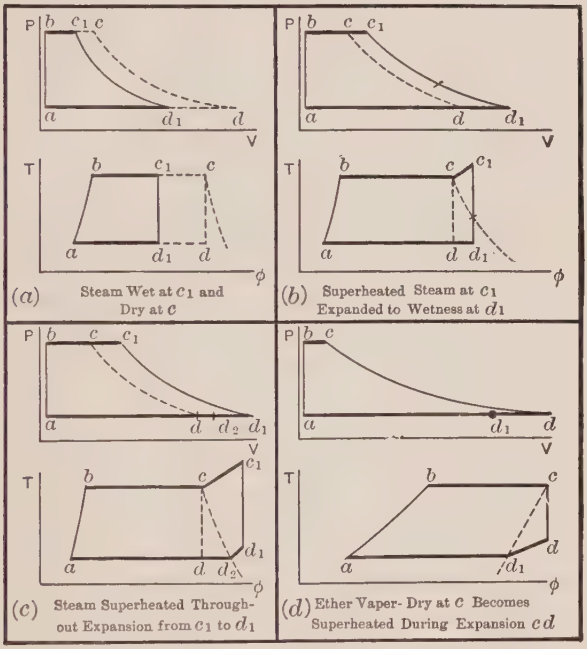

(a) Steam Wet at c_1 and Dry at c
(b) Superheated Steam at c_1 Expanded to Wetness at d_1
(c) Steam Superheated Throughout Expansion from c_1 to d_1
(d) Ether Vaper- Dry at c Becomes Superheated During Expansion cd

Fig. 58. —Clausius Cycles—Various Possibilities.

with the vapor of any material in either the saturated or superheated condition. The general equations for the Clausius cycle will be given in the latter part of this section. Some of the possible cases are shown in Fig. 58, in which the heavy lines in all instances represent constant pressures.

A word of explanation will probably help to make the construction of the diagrams in Fig. 58 clearer. In the Carnot cycle the upper and lower lines are defined as isothermals, while in the

Clausius cycle they are lines of constant pressure. For saturated vapors the two are the same, but for superheated vapors the two cycles present very different phenomena. The isobars give "horns" (at c_1) in the Tϕ-diagrams when in the superheated region, the height of these being determined directly by the degree of superheat.

(b) Another interesting difference results from the characteristics of this constant-pressure line. In the Clausius cycle the temperature rises during superheating, while in the Carnot it remains constant and the pressure drops. In the case of the former cycle, then, the hot body must have a temperature at least equal to that reached at the end of the superheating process and therefore higher than that of the working substance during the entire reception of heat. For this case, then, all the heat is received irreversibly.

(c) It thus develops that for all Clausius cycles the heat received along the line *ab* is received irreversibly, the hot body having a temperature at least as high as T_{c_1}, and for Clausius cycles in which superheating takes place, all the heat is received irreversibly, because the hot body must have a temperature at least as high as that attained by superheating. This cycle when using superheated vapor therefore departs still further from the criterion for maximum efficiency, and must have a theoretical efficiency lower than that of the same cycle with saturated vapor having the same maximum temperature. This conclusion is the more interesting because, notwithstanding the lower theoretical efficiency, real engines operating on this cycle obtain their highest commercial efficiency with superheated vapor. The reason for this will be brought out in a later chapter.

(d) For the Clausius cycle with the adiabatic expansion starting with *wet steam*, with quality x_1,

$$\Delta E = \frac{x_1 r_1}{T_1}(T_1 - T_2) + q_1 - q_2 - T_2(\Delta\phi_{l_1} - \Delta\phi_{l_2}), \quad (186)$$

and

$$Ef. = \frac{\Delta E}{x_1 r_1 + q_1 - q_2}. \quad . \quad . \quad . \quad . \quad . \quad (187)$$

(e) In the **general case** the quality (or temperature of superheat) at the end of the adiabatic expansion must first be found. This can be done by solving for x_d (or D_d) from

$$(\Delta\phi_l + x\Delta\phi_v + \Delta\phi_D)_d = (\Delta\phi_l + x\Delta\phi_v + \Delta\phi_D)_c, \quad . \quad . \quad (188)$$

or,
$$\left(\Delta\phi_l + x\Delta\phi_v + \overline{C}_p \log_e \frac{T_v + D}{T_v}\right)_d = \left(\Delta\phi_l + x\Delta\phi_v + \overline{C}_p \log_e \frac{T_v + D}{T_v}\right)_c . \quad (189)$$

If the steam is initially superheated $x_c = 1$; if wet, the entropy of superheat $(\Delta\phi_D)_c$ disappears. Should the value of x_d found be greater than 1.00, it indicates that the steam is still superheated, then D_d should be determined, using $x_d = 1$.*

Having determined x_d (or D_d), the work may be found from

$$\Delta E = (q + xr + \overline{C}_p D)_c - (q + xr + \overline{C}_p D)_d . \quad (190)$$

Also, the work may be found from

$$\Delta E = \Delta Q_1 - \Delta Q_2, \quad (191)$$

in which the values of ΔQ_1 and ΔQ_2, the heat supplied and the heat rejected, are equal respectively to the heat above 32° F., at the beginning and end of the isentropic expansion, and may be read directly from the Q-curves on the **Tϕ-chart** (Plate I, Appendix) or from the Q-scales on the **Mollier** or **Ellenwood Charts** (Plates II and IV in the Appendix).

The efficiency is

$$Ef. = \frac{\Delta E}{(q + xr + \overline{C}_p D)_c - q_2} \quad (192)$$

or

$$Ef. = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1 - q_2}. \quad (193)$$

This last form is the most convenient when the charts are used for obtaining ΔQ_1 and ΔQ_2. These heat quantities are of course measured above 32° F.

95. The Rankine Cycle. (a) This cycle is very similar to that last described, being obtained from it by a simple modification, the reason for which will be considered in a later chapter. The Rankine cycle, shown in Figs. 59 and 60 for dry steam at the beginning of expansion and superposed on the Clausius cycle for the same conditions, is seen to differ from the latter only in having the adiabatic expansion cut short by a constant-volume line *de*.

Since the adiabatic line is not continued to the lowest tempera-

* In solving for D_d it is necessary to assume a trial value of $\overline{C}_p$ and use the "cut and try method."

ture in the cycle, the expansion is said to be incomplete. As the figures show, the area of this cycle is less than that of the one

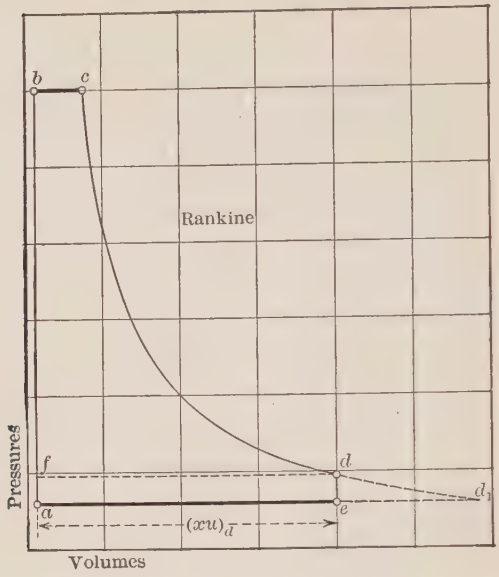


Fig. 59. — PV-Diagram for Rankine Cycle with Dry Saturated Water Vapor at *c*.

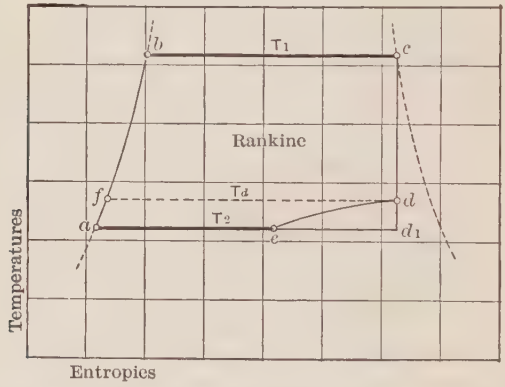


Fig. 60. — Tϕ-Diagram for Rankine Cycle with Dry Saturated Water Vapor at *c*.

having complete expansion, while the heat added along *ab* and *bc* is the same in both. It therefore follows that the Rankine cycle must be still less efficient than the Clausius. Despite this fact,

it is one of the most commonly used vapor cycles, being that approximated by most reciprocating steam engines.

(b) During the constant-volume pressure drop, *de*, heat is given up irreversibly by the working material because the cold body receiving that heat must have a temperature at least as low as T_e. Strictly interpreted, the line *de* on the Tϕ-diagram represents an equivalent reversible process.

It is evident that all the heat given to the cold body is not rejected when the working substance has the same temperature as that body, and hence this cycle should have a lower efficiency than a similar Clausius cycle. This has just been shown to be the case.

Work per Pound of Dry Saturated Steam Used in Rankine Cycle.

(c) With vapor dry and saturated at the beginning of adiabatic expansion, the work per pound is

$$\Delta E = \Delta Q_1 - \Delta Q_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (194)$$

$$\Delta E = \{(q_b - q_a) + r_1\} - \{(x_d\rho_d + q_d) - (x_e\rho_e + q_e) + x_e r_2\} \text{ B.t.u.} \quad . \quad . \quad (195)$$

In this expression the difference $(x_d\rho_d + q_d) - (x_e\rho_e + q_e)$ is the difference of intrinsic energy possessed by the vapor at the points *d* and *e*. It is obvious that, to decrease the pressure at constant volume, heat must be abstracted, and since no external work, positive or negative, is done, all heat removed must come from the stock of intrinsic heat energy possessed by the material at *d*. To use Eq. (195), however, the two qualities x_e and x_d must be determined first.

(d) A more useful expression may be developed as follows: Reference to Fig. 60 shows that the work of the cycle is represented by the sum of areas *fbcd* and *afde*. The former area is the same as that of a Clausius cycle with temperature limits T_1 and T_d, and its heat value can be computed from Eq. (177). The area *afde* corresponds to the similarly lettered area in the PV-diagram, Fig. 59, and hence represents $A(P_d - P_e) \cdot (xu)_d$ B.t.u. of work. Hence the work of the Rankine cycle for steam initially *dry* and saturated is

$$\Delta E = \{\lambda_1 - q_f - (xr)_d\} + A(P_d - P_e) \cdot (xu)_d, \quad . \quad (196)$$

and all quantities in this expression are either known at the outset or are obtainable directly from the Steam Table, with the exception of x_d, which can be obtained from Eq. (188) or (189).

Efficiency of Rankine Cycle Using Dry Saturated Steam.

(e) The heat received in this cycle is the same as that in the Clausius cycle, that is,

$$\Delta Q_1 = \lambda_1 - q_2.$$

Hence the efficiency is

$$Ef. = \frac{\Delta Q_1 - \Delta Q_2}{\Delta Q_1} = \frac{\Delta E \text{ from Eq. (196)}}{\lambda_1 - q_2} \quad . \quad . \quad . \quad (197)$$

96. The Rankine Cycle in General. (a) Starting with steam initially **wet,** the work done is

$$\Delta E = \{(x_1 r_1 + q_1 - q_f) - x_d r_d\} + A(P_d - P_2)\, x_d u_d, \quad . \quad (198)$$

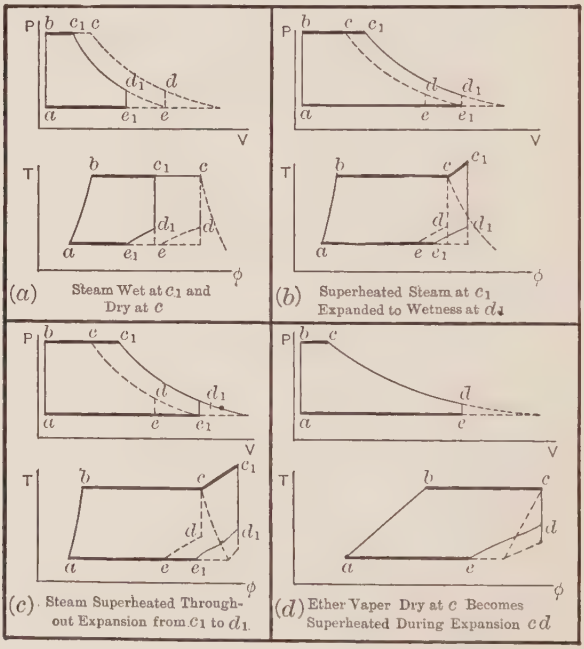

(a) Steam Wet at c_1 and Dry at c
(b) Superheated Steam at c_1 Expanded to Wetness at d_1
(c) Steam Superheated Throughout Expansion from c_1 to d_1
(d) Ether Vaper Dry at c Becomes Superheated During Expansion cd

Fig. 61. — Rankine Cycles — Various Cases.

in which all quantities are known or are obtainable from the Steam Tables except x_d, which must be computed by using Eq. (188) or (189).

(b) In the most **general case,** having first determined from

the equation last mentioned the quality x_d (or superheat D_d) at the end of the adiabatic expansion, the work done is

$$\Delta E = \{(q + xr + \overline{C}_p D)_c - q_f - (xr + \overline{C}_p D)_d\} + A(P_d - P_2)\,V_d, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (199)$$

where $V_d = (x\mathbf{V} - 0.017)_d$ if the steam is wet at d, or $V_d = (\mathbf{V}_s - 0.017)_d$ if superheated. $\mathbf{V}_s$ can be found from Eq. (134).

$$Ef. = \frac{\Delta E}{(q + xr + \overline{C}_p D)_c - q_2}. \quad . \quad . \quad . \quad (200)$$

(c) As in the other vapor cycles, there are a number of different possibilities as regards the working substance, but every case can be worked out more or less simply by means of the expressions already developed. Various cases of the Rankine cycle are shown in Fig. 61.

(d) The Rankine cycle can be solved readily by the use of **Ellenwood's Charts** (Plates III and IV, Appendix), since these have one coördinate representing volumes. Thus, letting Fig. 59 represent the Rankine cycle in general, it is evident that this cycle may be considered as composed of the Clausius cycle *fbcd* and the rectangular area *afde*. Then, if ΔQ_c and ΔQ_d are respectively the total heats at the beginning and end of the isentropic expansion, as obtained from Plate IV,* the

$$\text{Net work of area } fbcd = \Delta Q_c - \Delta Q_d \text{ B.t.u.}$$

and, if the constant-pressure external work represented by the area below *fd* is ΔE_d, per pound, and if that below *ae* is ΔE_e, then

$$\text{Net work of rectangle } afde = \Delta E_d - \Delta E_e \text{ B.t.u.}$$

in which the values of ΔE_d and ΔE_e can be obtained directly from Plate III for the volume V_{de} and the pressures (or qualities or superheats) already given or determined for d and e.

Then for the Rankine cycle (per lb. of working substance)

$$\text{Net work} = (\Delta Q_c - \Delta Q_d) + (\Delta E_d - \Delta E_e) \text{ B.t.u.}$$

Ellenwood's Charts offer an easy solution for this cycle regardless of whether it is the pressure or the volume at d that is initially known. Without these charts a laborious cut-and-try process must be used if only the volume is given.

* When ΔQ_d is obtained from Plate IV the values of the volume V_{de} and pressure at d should be noted as they will be needed later in obtaining ΔE_d from Plate III.

97. Cycle with Rectangular PV-Diagram. (a) This cycle is the least efficient of all the vapor cycles in practical use. It is

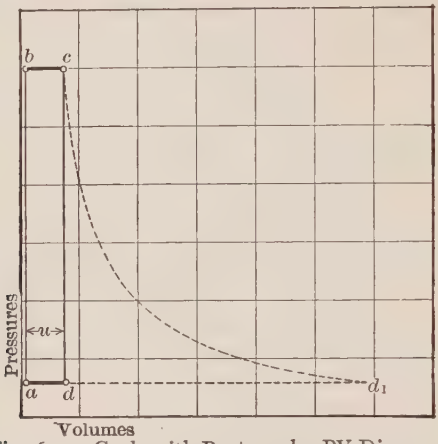


Fig. 62. — Cycle with Rectangular PV-Diagram.

composed of two constant-pressure lines joined by two lines of constant volume, as shown in the PV-diagram, Fig. 62, and in the Tϕ-diagram, Fig. 63.

The diagrams show this cycle superimposed upon a Clausius

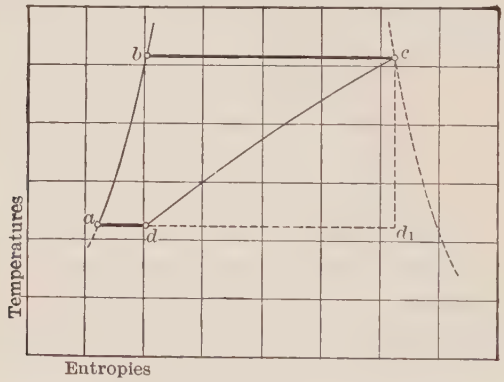


Fig. 63. — Tϕ-Diagram for Rectangular PV Cycle.

cycle so chosen that the same weight of working substance is used in each. It is evident that the Clausius cycle will require a much larger cylinder than the cycle under consideration, but the work per cycle will also be much greater per pound of vapor.

The Tϕ-diagram shows that the heat absorbed is the same with both cycles, namely, the area beneath the line *abc*. The work done is, however, greater with the Clausius cycle than with the rectangular PV cycle, as is shown by the inclosed areas of the diagrams. It follows that the efficiency of the cycle with rectangular PV-diagram must be less than that of the Clausius cycle. The Rankine cycle for the same heat input evidently gives an amount of external work intermediate between that obtained with the Clausius cycle and that obtained with the cycle under consideration, and must therefore have an intermediate efficiency. The rectangular PV-diagram may be looked upon as a limiting case of the Rankine cycle, the Clausius cycle being the other limit.

Work per Pound of Dry Saturated Steam Used.

(**b**) From Fig. 62 it is apparent that

$$\Delta E = A\,(P_1 - P_2)\,u. \quad . \quad . \quad . \quad . \quad (201)$$

Efficiency of the Cycle Using Dry Saturated Steam.

(**c**) The heat received is the same as that in the Clausius cycle. Hence

$$Ef. = \frac{\Delta E}{\Delta Q_1} = \frac{A\,(P_1 - P_2)u}{(\lambda_1 - q_2)}. \quad . \quad . \quad . \quad (202)$$

98. The Rectangular PV Cycle in General. In any case

$$\Delta E = A\,(P_1 - P_2)\,x_c u, \quad . \quad . \quad . \quad . \quad . \quad (203)$$

where $x_c u = (x\mathbf{V} - 0.017)_c$ if the steam is wet, or $= (\mathbf{V}_s - 0.017)_c$ if superheated. $\mathbf{V}_s$ can be found from Eq. (134). The general expression for the efficiency is

$$Ef. = \frac{\Delta E}{(q + xr + \overline{C}_p D)_c - q_2}. \quad . \quad . \quad . \quad (204)$$

For method of using Ellenwood's Charts for solving the Rectangular PV cycle see the middle of **96**(**d**).

CHAPTER XIII.

POWER, EFFICIENCY, AND PERFORMANCE.

CERTAIN general definitions which are necessary in the consideration of real engines are collected in this chapter. They will be discussed here very briefly; most of them will be considered more fully in later chapters and some belong more properly to the province of Experimental Engineering.

99. Power. (a) In English-speaking countries, the **foot-pound** (ft.-lb.) is the unit of work generally used by engineers. The unit of power, or unit of the "rate" of doing work, is the **horse power** (h.p.); it equals the power equivalent to the doing of 33,000 foot-pounds of work per minute.

Then the horse power developed by any apparatus is

$$\text{h.p.} = \frac{\textit{Total ft.-lbs. of work developed per min.}}{33{,}000} \quad . \quad (205)$$

The heat equivalent of one horse power is

$$\text{One h.p.} = 33{,}000 \div 778 = 42.42 \text{ B.t.u. per min.}^* \quad (206)$$

(b) If work is done for one hour at the rate of one horse power, the total work done is called one **horse-power hour** (h.p.-hr.).

Then, one h.p.-hr. $= 33{,}000 \times 60 = 1{,}980{,}000$ ft.-lbs. . (207)

$$= 1{,}980{,}000 \div 778 = 2545 \text{ B.t.u.}^* \quad . \quad (208)$$

100. Distinction between Real and Ideal Engines. In considering the ideal or thermodynamic engine in preceding chapters, a working substance was assumed to pass through cycles within a closed cylinder, and it was found that a certain amount of work, ΔE, would be delivered to the piston during each cycle. The material of the cylinder and piston was assumed to have certain properties which no available material really has. The cylinder and piston were assumed to neither absorb nor conduct

* The A.S.M.E. Power Test Code (1915) uses 777.54 ft.-lb. as the equivalent of one B.t.u. Then one h.p. = 42.44 B.t.u. per minute or 2546.5 B.t.u. per hr.

heat. The piston was supposed to be without leakage and friction; and any other necessary mechanism of the engine was assumed frictionless. These conditions cannot be realized in practice. Therefore, the action of a real engine must differ considerably from the conceived action of an ideal engine.

Losses in real heat engines may be classified as follows:

(1) **Cycle loss,** — for even with the ideal cycle only part of the heat supplied can be converted into work.

(2) **Cylinder losses,** or those which occur within the real cylinder because the ideal cycle is not perfectly produced. These losses reduce the work actually delivered to the piston by the working substance.

(3) **Friction losses,** occurring in the mechanism used in the transmission of work between the piston face and the place of utilization.

101. The Indicator. The work actually performed on the piston by the working substance in the cylinder of the real engine and the pressure-volume changes that actually occur within the cylinder can be determined by using the instrument called the " Indicator," which can be made to draw the PV-diagram for the changes actually occurring. The comparison of such a diagram with the ideal one aids in determining the cylinder losses.

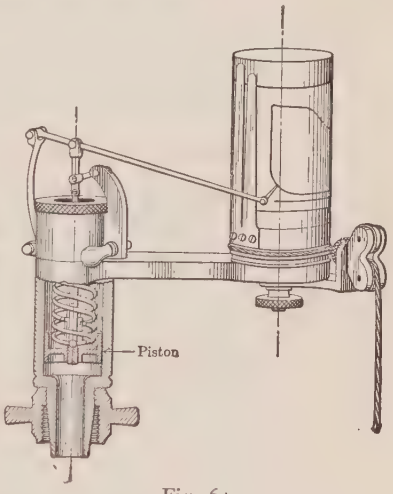


Fig. 64.

This instrument is shown in Fig. 64. A card is mounted on the outside of a metallic cylinder which is caused to oscillate in unison with the motion of the engine piston. A pencil, which may be pressed against this card, is actuated by a small, spring-balanced piston subjected to the same pressure as the engine piston. Thus the card movement is proportional to the volume displaced by the engine piston, while the pencil movement is pro-

portional to the pressure which actuates the piston. The pencil movement is at right angles to the card movement, and hence a pressure-volume diagram with rectangular coördinates, such as *abcde* in Fig. 65, is drawn. If the card cylinder oscillates under the pencil while the indicator piston is disconnected from the engine cylinder and subjected to atmospheric pressure, a horizontal line, called the **atmospheric line,** will be drawn.

102. The Indicator Diagram. (a.) In the pressure-volume diagram drawn by the indicator, as in the PV-diagrams previously considered, the inclosed area represents the work done upon the engine piston by the working substance during the cycle.

(b) The **Pressure Scale,** S_p, is commonly expressed as the number of pounds per square *inch* of piston area, corresponding to one inch of ordinate. This is also called the "Spring Scale." Obviously, the scale giving pressure per square *foot* of piston area is $144 \times S_p$.

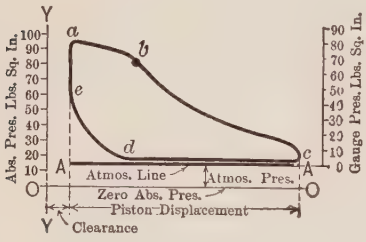


Fig. 65.

The datum of absolute pressures is a horizontal line, *OO* in Fig. 65, drawn at a distance below the atmospheric line, *AA*, equal to the atmospheric pressure as measured on the pressure scale. Thus for any point, the absolute intensity of pressure per square inch of piston area = (ordinate above *OO*) $\times S_p$; similarly, the pressure above atmospheric = (ordinate above *AA*) $\times S_p$. The latter pressure is usually called the "*gauge pressure.*"

(c) The **Volume Scale,** S_v, is the displacement, in cubic feet, per square *foot* of piston, represented by one inch of abscissa.

The datum of total volumes is a vertical line, *YY* in Fig. 65, located to the left of *Aa* at a distance representing, to scale, the "*clearance volume*" or space in the cylinder occupied by the working substance when the piston is at the beginning of its stroke. Thus for any point on the diagram, the total volume of working substance in the cylinder equals $S_v \times$ (abscissa from *YY*), and the volume displaced by the piston is $S_v \times$ (abscissa from *Aa*).

(d) The **Scale of Work,** S_w, corresponding to one square inch of area of the diagram = $S_w = S_p \times S_v$ foot-pounds per square *inch* of piston area, or 144 S_w ft.-lbs. per square *foot*.

The work done per cycle by the working substance acting on the total piston area, as represented by the area of the diagram, is called the **Indicated Work.** Thus the i.w. = (area of diagram) $\times$ (area of piston in sq. in.) $\times S_w$.

The corresponding rate of doing indicated work is expressed in terms of horse power; it is called the **Indicated Horse Power** (i.h.p.) and is computed by using Eq. (205).

(e) Consider Fig. 65 as an actual diagram taken from an engine. From the point *a* the engine piston moved out until the point *c* was reached. By virtue of the property of the PV-diagram, the area between the lines *abc* and *OO* represents the work done *upon* the piston by the expanding working substance. This work may be computed by multiplying the average pressure on the face of the piston of the engine by the piston's movement. To find the average pressure per square *inch* of piston, divide the square inches of area between *abc* and the pressure datum *OO* by the length *AA* in inches and multiply this average height by S_p. Multiplying this mean intensity of pressure by the area of the piston in square inches and by the length of stroke of the piston in feet gives the work done during the out stroke of the piston.

(f) Similarly, the area under the line *cde* represents the work done by the piston upon the working substance during the return stroke, and the mean ordinate of this area multiplied by S_p gives the average intensity of pressure against which the engine piston moved during this stroke. This, multiplied by the area of the piston in square inches and by the stroke of the piston in feet, gives the work in foot-pounds done by the piston *on* the working substance during the return stroke.

(g) The useful work delivered to the piston during one cycle equals the difference between the work done upon it on the out stroke and that done by it on the working substance during the return stroke.

The amount of work actually accomplished would have been the same if the difference between these two average pressures had acted upon the piston during one stroke only. The value of this difference is, however, given by dividing the area *abcde* by

the length of the diagram and multiplying by S_p. This is known as the **mean effective pressure** (m.e.p.), and is defined as the pressure which, operating on the face of the piston during one stroke, would do the same amount of work as is actually done per cycle by the variable pressure really acting.

(h) In terms of the mean effective pressure, which will hereafter be designated by p_m, the work done upon the piston by the working substance, per cycle, is

$$\text{Work} = p_m \cdot a \cdot L \text{ ft.-lbs.}, \quad . \quad . \quad . \quad . \quad (209)$$

in which a represents the area of the engine piston in square *inches* and L is the stroke in *feet*.

If there are n cycles per minute, the work per minute will be n times the work per cycle, and the **indicated horse power** of the engine will be

$$\text{i.h.p.} = \frac{p_m L a n}{33{,}000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (210)$$

(i) Eq. (210) can be used to determine the diameter of cylinder needed to develop any i.h.p., provided the m.e.p., the length of stroke, and the number of cycles per minute are known. Thus the effective **area of the piston** must be

$$a = \frac{33{,}000 \text{ i.h.p.}}{p_m L n}, \quad . \quad . \quad . \quad . \quad . \quad (211)$$

from which the piston diameter follows.

103. Methods of Determining the Area of an Indicator Diagram. (a) The area of an indicator diagram can be determined (1) by placing transparent "cross-section paper" over the diagram and counting the squares surrounded; (2) by using some such form of mechanical integrating instrument as the "planimeter;" (3) by applying the "method of ordinates;" or (4) by using some integration rule such as the "Trapezoidal Rule"* and "Simpson's One-third Rule."†

(b) One form of **planimeter** is shown in Fig. 66. It consists of two arms jointed together, one terminating in a "fixed point" which is a stationary pivot, while the other carries a "tracing point." The third support for the instrument is a point of the

* For this rule see Kent's "Pocket Book."

† See Church's "Notes on Mechanics" or Kent's "Pocket Book," published by John Wiley & Sons.

rim of a graduated wheel or "record roller." If the record wheel is set at zero and the tracing point is moved clockwise around the outline of the diagram and is returned to its original position, the area of the figure is given by the reading of the record wheel. The theory and use of planimeters is treated in books on Experimental Engineering.* The mean ordinate is found by dividing the area by the length of diagram, and the m.e.p. is the product of the mean ordinate and the pressure scale.

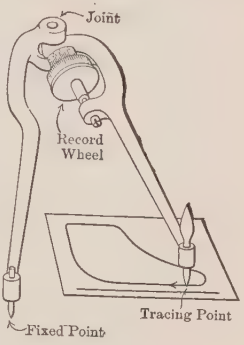


Fig. 66.

(c) In the **method of ordinates,** the length of the diagram is divided into a number of equal parts, with interval Δx as in Fig. 67, and ordinates are drawn, as 1, 2, 3, etc., in the figure. Central intermediate ordinates are then drawn and the intercepts y_1, y_2, y_3, etc., are scaled and used as the mean heights of the elementary areas between ordinates. The area of the diagram is approximately $A = \Sigma y \times \Delta x$, and the mean ordinate is $y_m = \dfrac{\Sigma y}{(no.\ of\ ordinates)}$.

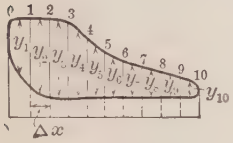


Fig. 67.

This method is not strictly correct, for the middle intercepts are not necessarily the mean heights of the elementary areas. These mean heights can be found quite accurately by the method shown in Fig. 68. Here lines AB and CD (not necessarily horizontal) are so drawn that areas a_1 and a_2 are equal and that $b_1 = b_2$. Then the distance y between the centers of these lines is the true mean height. The equality between areas a_1 and a_2 and between b_1 and b_2 can be estimated very accurately by eye.

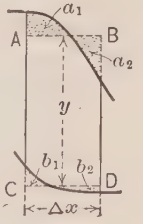


Fig. 68.

104. Delivered Power. (a) In Section 100 it was stated that only a part of the net work done on the piston by the working substance is delivered by the engine, as there is a friction loss in the moving parts. The power which actually is made available by the engine is variously called the

* See Carpenter and Diederich's "Experimental Engineering," published by John Wiley & Sons.

delivered horse power (d.h.p.), the **brake horse power** (b.h.p.), and the **effective horse power** (e.h.p.).

The difference between the indicated horse power and the delivered horse power is a measure of the power lost in friction, and is called the **friction horse power** (f.h.p.). Then

$$\text{f.h.p.} = \text{i.h.p.} - \text{d.h.p.} \quad . \quad . \quad . \quad . \quad . \quad (212)$$

The indicated horse power can be determined by means of the indicator, and hence, if either the friction horse power or the delivered horse power can be measured, all three of the quantities of Eq. (212) become known.

(**b**) The direct measurement of the friction horse power is usually impossible, but several approximate methods are used. One scheme depends upon the assumption that the power consumed in engine friction is constant for all values of delivered power. This assumption is not accurate, but may be used for approximation. Assuming it true, the indicated horse power obtained when the engine is running at speed with no external load, that is, when all the indicated power is applied to overcome friction, may be taken as a measure of the friction horse power.

Sometimes it is possible to drive an engine at its normal speed, from some external source of power, such as an electric motor or a shaft. When this can be done, and when the power thus consumed can be measured, it furnishes an approximate determination of the friction horse power. However, it is necessary to make the same assumption as in the previous case.

The usual method is to determine the delivered horse power experimentally and to calculate the friction horse power by Eq. (212).

The delivered horse power may be measured by the use of a prony brake or similar absorption or transmission dynamometer; hence the term "brake horse power." For large engines absorption dynamometers become elaborate and expensive and are seldom used except in special cases.

105. Efficiencies. (**a**) Efficiency is the ratio of result to effort. For the heat engine there are several such ratios, which depend upon the meanings given to the terms "result" and "effort." They are useful in comparing performances of different engines, in locating losses, and in showing opportunities for improvement. Unfortunately, there is lack of uniformity in the

names applied to the various efficiencies, and in some cases the same term has been used for entirely different ratios. In the following discussion the names which are apparently the most suitable have been adopted.

Fig. 69 is a diagram showing the energy stream. Here, as in Fig. 3, the width of stream shows the amount of energy still available for doing external work. As the stream progresses losses occur, as shown by the offshoots, and less energy remains

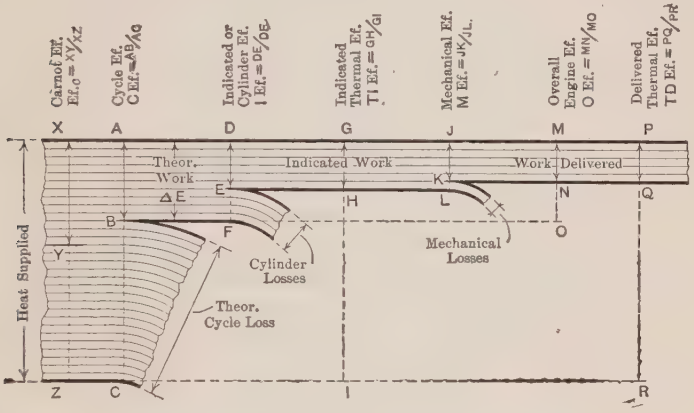


Fig. 69.

available for doing external work. The several efficiencies, which will now be considered, may be studied in connection with this figure, and the relation between the various ones will be made clearer by referring to the figure as the discussion progresses.

(b) **Carnot Efficiency.** It has been shown that the efficiency of the **Carnot** cycle, and of all other reversible cycles, is the theoretical maximum with any given temperature limits. It is an ideal efficiency, but is impossible of attainment in any real case. Its value regardless of the kind of working substance is

$$Ef_c = \frac{T_1 - T_2}{T_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (213)$$

In Fig. 69, XZ represents the heat supplied and XY that which would be delivered as external work if the Carnot cycle were followed; hence the Carnot Efficiency is $Ef_c = \frac{XY}{XZ}$.

(c) **Cycle Efficiency.** In all real engines the working substance in its action approximates one of the theoretical cycles already developed. Each of these cycles was shown to have inherent thermodynamic loss and a theoretical efficiency less than unity. This efficiency will hereafter be called the Cycle Efficiency, *CEf*. It is shown in Fig. 69 by the ratio $\frac{AB}{AC}$.*

For example, if a steam engine is assumed to follow the ideal Clausius cycle, the Cycle Efficiency is given by Eq. (192), and the work, ΔE_1, per pound of material by Eqs. (188) to (191). In Fig. 69, AB represents ΔE_1.

No real engine actually attains the efficiency of its theoretical cycle because of unpreventable losses, but the Cycle Efficiency represents the best result attainable with the cycle in an engine having no extra-thermodynamic losses.

(d) **Relative Efficiency.** It would seem that the engineer should be able to design and construct engines to operate with the Carnot and other reversible cycles, and thus approximate the ideal efficiency. However, practical reasons generally compel the use of engines approximating theoretical cycles that are thermodynamically less efficient. This reduces the possible efficiency even before the practical losses are considered.

A measure of this reduction is obtained by dividing the Cycle Efficiency of the engine considered by the Carnot Efficiency. The quotient will be called the Relative Efficiency, *REf*, and is

$$REf = \frac{CEf}{Ef_c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (214)$$

Referring to Fig. 69, it is evident that

$$REf = \left(\frac{AB}{AC}\right) \div \left(\frac{XY}{XZ}\right) = \frac{AB}{XY}$$

$$= \frac{\text{Work done by cycle under consideration}}{\text{Work done by Carnot cycle}}.$$

(e) **Indicated or Cylinder Efficiency.** In actual engines, as stated, the work done upon the piston by the working substance is of course always less than the theoretical quantity; that is,

* Note that any engine operating on a cycle which is theoretically *reversible* will have a Cycle Efficiency equal to the Carnot Efficiency (as in (b)). In other cases the amount by which the *CEf* falls short of Ef_c indicates the theoretical disadvantage of the irreversible cycle.

it is less than the product of the Cycle Efficiency by the heat supplied.

The ratio of work actually done to work theoretically possible measures the perfection of design, construction, and operation of the cylinder, piston, and valves.

This ratio, which will be called either the Indicated or the Cylinder Efficiency,* IEf, can be expressed in several ways as follows:

$$IEf = \frac{\text{Area of actual indicator diagram}}{\text{Area of theoretical PV-diagram}}. \quad . \quad . \quad . \quad . \quad (215a)$$

$$= \frac{\text{Indicated work per pound of working substance}}{778\ \Delta E \text{ (for corresponding theoretical cycle)}} \quad (215b)$$

$$= \frac{\text{Heat utilized per pound of working substance}}{\Delta E \text{ (for corresponding theoretical cycle)}}. \quad (215c)$$

$$= \frac{\text{I.h.p.}}{\text{Theoretical horse power}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (215d)$$

In the energy stream shown in Fig. 69, DE represents the indicated work and AB the theoretical work. Evidently the Cylinder Efficiency is

$$IEf = \frac{DE}{AB} = \frac{DE}{DF}.$$

For example, if, in the case of the steam engine previously cited, the work per pound of steam shown by the actual indicator diagram is $\Delta E'$, and if ΔE_1 is the work with the Clausius cycle, then the $IEf = \frac{\Delta E'}{\Delta E_1}$, and DE in Fig. 69 represents $\Delta E'$.

(f) **Mechanical Efficiency.** The ratio of work delivered by the engine to work received by the piston (equal to the ratio of delivered power to indicated power) is called the Mechanical Efficiency, MEf. Thus

$$MEf = \text{d.h.p.} \div \text{i.h.p.} \quad . \quad . \quad . \quad . \quad . \quad (216)$$

This fraction gives the proportion of the power received by the piston which actually becomes available as mechanical power for the consumer. The loss is a mechanical one due to friction of the mechanism.

* In the case of steam engines the A.S.M.E. Power Test Code (1915) calls this the "**Rankine Cycle Ratio referred to the i.h.p.**" and assumes that the cycle followed is the Clausius or Rankine Complete Expansion Cycle. Many authors term this the "**Efficiency Ratio,**" which is recommended by the British Inst. C. E., Vol. CXXXIV, p. 796.

In Fig. 69, JK represents the energy delivered by the engine, and DE, or JL, shows the indicated work done on the piston; hence the mechanical efficiency is $\frac{JK}{JL}$.

(g) **Thermal Efficiency on the I.h.p.** The ratio of indicated work done (GH in Fig. 69) to heat supplied in the working substance (XZ or GI) is useful in showing the combined efficiency of the cycle and the cylinder with appurtenances. It will be called the Thermal Efficiency on the i.h.p., abbreviated $TIEf$, and is

$$TIEf = \frac{\text{Indicated work in B.t.u.}}{\text{Heat supplied to cylinder}}. \quad . \quad . \quad (217)$$

Obviously, this efficiency equals the product of the Cycle Efficiency by the Indicated Efficiency, that is,

$$TIEf = CEf \times IEf. \quad . \quad . \quad . \quad . \quad . \quad (218)$$

The $TIEf$ is shown in Fig. 69 by the ratio $\frac{GH}{GI}$.

(h) **Thermal Efficiency on the Brake or Delivered Power.** The ratio of delivered work (PQ in Fig. 69) to the heat supplied the engine will be called the Thermal Efficiency on the Brake or Delivered Power, $TDEf$. Thus

$$TDEf = \frac{\text{Work delivered in B.t.u.}}{\text{Heat supplied cylinder}}. \quad . \quad . \quad (219)$$

Also, it is evident that

$$TDEf = TIEf \times MEf. \quad . \quad . \quad . \quad . \quad (220)$$

The $TDEf$ is shown in Fig. 69 by the ratio $\frac{PQ}{PR}$.

(i) **The Overall Efficiency of the Engine.** The true efficiency of the engine *as a whole*, compared with the ideal or thermodynamic engine with the same cycle, will be called the Overall Efficiency, OEf. This takes account of both the cylinder and the mechanical losses.* Hence

$$OEf = IEf \times MEf. \quad . \quad . \quad . \quad . \quad . \quad (221)$$

The OEf in Fig. 69 is the ratio MN/AB or MN/MO.

A study of Fig. 69 shows that all of these efficiencies follow one another in logical order, and that each has a definite bearing upon the analysis of the performance of real engines.

* In the case of steam engines the A.S.M.E. Power Test Code (1915) calls this the "**Rankine Cycle Ratio referred to the br. h.p.**"

106. Engine Performance. (a) The relative performance of two heat engines can be determined by comparison of the amounts of heat used to produce a given amount of work. The unit of work usually adopted for comparison is either the indicated horse-power hour or the delivered horse-power hour. Thus the **Rate of Heat Consumption** may be defined as B.t.u. required per horse-power hour, that is, $\frac{\text{B.t.u.}}{\text{i.h.p.-hr.}}$ or the $\frac{\text{B.t.u.}}{\text{d.h.p.-hr.}}$, as the case may be.

(b) If the amount of working substance used per hour is weighed and if the h.p. is determined, then the weight of material per h.p.-hr. can be computed. Evidently, if W_i, or W_d, is the weight of working substance per h.p.-hr., and if ΔQ is the heat per pound of material, then

$$\frac{\text{B.t.u.}}{\text{i.h.p.-hr.}} = W_i \times \Delta Q, \quad . \quad . \quad . \quad . \quad . \quad (222)$$

$$\frac{\text{B.t.u.}}{\text{d.h.p.-hr.}} = W_d \times \Delta Q. \quad . \quad . \quad . \quad . \quad . \quad (223)$$

Since the equivalent of one h.p.-hr. is 2545 B.t.u., and since the Thermal Efficiency is the ratio of the work actually done to the heat supplied, it is evident that

$$\frac{\text{B.t.u.}}{\text{h.p.-hr.}} = \frac{2545}{TIEf, \text{ or } TDEf, \text{ as the case may be}}. \quad . \quad . \quad (224)$$

If several engines use working substances of the same kind with the same heat content per pound, the relative performances can be found by comparing the **Rates of Consumption of Working Substance** (i.e., pounds per i.h.p.-hr. or per d.h.p.-hr.). These values are known as **Engine Economies.**

Further, if unit weights of these working substances receive their store of heat from equal weights of fuel, the **Rates of Fuel Consumption** (pounds per i.h.p.-hr. or d.h.p.-hr.) may be used for comparison.

(c) Graphical representations of engine performances are often very useful. They may be based upon the scheme shown in Fig. 70, which applies to an impossible machine supposed to convert into mechanical energy all of the heat supplied it; — thus it is the **case with efficiency of 100 per cent.**

Since 2545 B.t.u. are equivalent to one h.p.-hr., and since in this case the efficiency is the same at all rates of power develop-

ment, that is, at all "loads," the curve showing the Rate of Heat Consumption, or R-curve, is a horizontal line with ordinate 2545 B.t.u., as shown in the figure. The scale for this line is at the right.

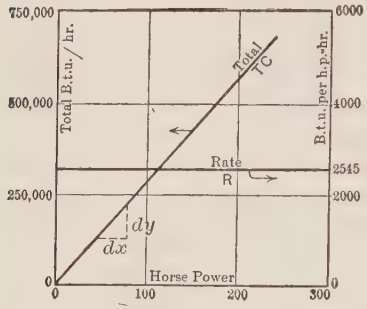


Fig. 70.

The Total Heat Consumption per hour at any load is the product of the horse power and the corresponding rate. Thus the curve showing the Total Consumption, or the TC-curve, results from plotting the products of corresponding abscissas and ordinates of the R-curve. Since the latter is a horizontal line in this case, the corresponding TC-curve must be a straight line passing through the origin and with slope corresponding to the rate. The scale for this curve is given at the left of the figure.

(d) When the B.t.u. per pound of working substance remains constant, it is sometimes convenient to construct curves similar to those in Fig. 70 but for the consumption of the working substance instead of B.t.u. Thus the R-curve would represent the Rate of Consumption of Working Substance (as pounds of steam per h.p.-hr., or cubic feet of gas per d.h.p.-hr., etc.), and the TC-curve would represent the total consumption of working substance (as total weight of steam or cubic feet of gas per hour).

Sometimes similar curves are drawn to represent the rate and total consumption of fuel used (as pounds of coal per h.p.-hr., and total weight per hour).

(e) According to assumption the efficiency of this impossible device is constant, and if the efficiency line were drawn it would be horizontal at a height corresponding to 100 per cent. Even in the best theoretical cycles, that is, the Carnot, and the other reversible ones, the work performed is very much less than the mechanical equivalent of the heat supplied, and the efficiency is always much less than unity.

(f) In the **real engine** the efficiency, and hence the rate, instead of being constant, varies characteristically with the load; thus, instead of being straight, as in Fig. 70, the lines representing the efficiency and rate may be curved, as is shown for one real

engine in Fig. 71. Further, the TC-curve will not pass through the origin of coördinates, but will have a positive intercept on the Y-axis, as shown in Fig. 71. This is because there is a heat loss when the external load equals zero; for, even when an engine is running without delivering power, there is heat lost in radiation and conduction and in overcoming friction, and if the engine is motionless at the operating temperature, there is still the loss due to radiation and conduction.

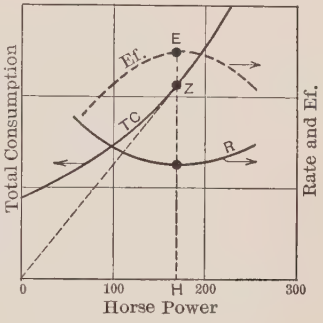


Fig. 71.

(g) The ordinate scales in Fig. 71, as in the case of Fig. 70, may be made to read in thermal units, pounds of working substance or pounds of fuel.

The ratios of the number 2545 to the different ordinates of the heat-rate curve evidently give values of the Thermal Efficiencies at different loads, as shown by the curve *Ef* in Fig. 71. This curve will give either the *TIEf* or the *TDEf*, according to the basis used in determining the R-curve. Also, the Thermal Efficiencies are given by the ratios of ordinates in Fig. 70 to the corresponding ones in Fig. 71.

Similar comparisons between the curves for any theoretical cycle with those for the Carnot cycle will give the Relative Efficiencies for the former.

(h) In Fig. 71 a dotted line is drawn from the origin tangent to the TC-curve. The point of tangency, *Z*, determines the abscissa, or horse-power output, at which the efficiency is maximum and the rate minimum. Evidently, the best economy is obtained when the engine develops this power, and, other things being equal, an engine should be of such size as to operate most of the time at or near this load. If the engine normally furnishes more or less than this power, it is either too large or too small *from the standpoint of economy only*. It will appear later, however, that many other considerations enter into the choice of size of engine best suited for a given set of conditions.

CHAPTER XIV.

THE THEORETICAL STEAM ENGINE.

107. General. (a) In the actual steam engine only a portion of the heat supplied in generating the steam is converted into useful work. This portion at maximum is only about 25 per cent and ordinarily is from 5 to 12 per cent. All the rest of the heat, from 75 to 95 per cent, is lost, and represents a proportionate waste of fuel and of money spent for it. It is very important for one who is to be connected with steam engineering to understand why this great loss occurs and how it can be minimized.

(b) The greater part of the heat loss would occur even in the theoretically perfect engine, — because of imperfections inherent in the ideal cycle; and the exact extent of the loss in this case can be readily computed. The further losses that occur in the actual engine are due to physical imperfections; and their amounts can be determined experimentally, while their causes and proportionate distribution can be studied by comparing the actual cycle with the ideal. Many of the losses can be determined by comparing the actual cycle with the ideal ones, — the Carnot, Clausius, and Rankine.

108. The Carnot Cycle and the Steam Engine. (a) As the Carnot cycle (Section 91) gives the greatest possible efficiency, it would seem to be the most desirable cycle to use in any type of engine.

Heretofore, in discussing this cycle, it was assumed that all operations were performed within a single nonconducting cylinder, to the end of which could be attached the hot body, or the cold body, or the nonconducting head, as required during the cycle. While such an arrangement is conceivable, it cannot be realized materially, and to obtain an apparatus of practical value it is necessary that some parts of the cycle shall be per-

formed outside of the cylinder. In this latter case, however, the result will be the same, provided the cycle is carried through in the same manner as before. Thus the cycle may be performed in the following apparatus:

(b) Let the cylinder, the cylinder end, and the piston be perfectly nonconducting, and let the cylinder end be permanently attached. Then, instead of a hot body, let there be a pipe with a valve ("Steam Valve") connecting the cylinder to a boiler which will supply steam (heat) at the constant temperature T_1, corresponding to pressure p_1; and in place of the cold body let there be another pipe with a valve ("Exhaust Valve") connecting the cylinder to the condenser, in which the temperature is maintained constantly at T_2, corresponding to the exhaust pressure p_2. Such an arrangement, with the addition of a feed pump to return the condensate from the condenser to the boiler, completes the apparatus, which contains the simple elements of a steam power plant.

(c) In performing the Carnot cycle, note that (see Section 53):

(1) All heat from the external source must be received at the constant temperature T_1 of the source.

(2) All heat discharged to the cold body must be rejected at the constant temperature T_2 of the cold body. Hence:

(3) Before heat is received at the upper temperature T_1, the working substance must be brought to that temperature without receiving heat energy from the outside; so it must be done by adiabatic compression from T_2 to T_1, and

(4) Before heat is rejected, the temperature must be lowered from T_1 to T_2 without losing heat as such to the outside; so this must be accomplished by adiabatic expansion.

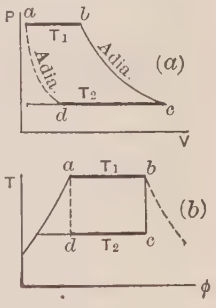


Fig. 72.

Referring to Fig. 72 (*a*) for the PV-diagram and to Fig. 72 (*b*) for the Tϕ-diagram (the two figures being lettered alike), the cycle would be performed in the following manner:

(d) **Isothermal Expansion** (line *ab*). Since in the Carnot cycle the working substance must receive all its heat from the outside source at the upper temperature T_1, the cycle must begin with water (say one pound) which has already been raised to this temperature. In the first operation, — starting with the

steam valve open, the exhaust valve closed, and piston against the cylinder head, — the boiler will supply the latent heat (at a constant temperature) to form the vapor, which will then occupy the volume V_b, the piston meanwhile moving out until it has swept through a volume equal to this. This first part of the cycle constitutes the period of **admission.** The steam valve is then closed, and the steam supply is "**cut off**" at point b.

(e) **Adiabatic Expansion** (bc in the figures). The steam is then allowed to expand adiabatically from b to c, within the cylinder, doing external work by moving the piston against a resistance until the temperature T_2 of the condenser is reached. The pressure is then p_2, the volume is V_c, and the piston is at the end of its stroke. This completes the second part of the cycle. The exhaust valve is then opened to "**release**" the steam from the cylinder, allowing it to flow to the condenser.

(f) **Isothermal Compression** (cd). On the return stroke the piston drives the steam out of the cylinder into the condenser, where, by the abstraction of heat, it is liquefied. During this operation the temperature remains constant at T_2, so the heat is rejected to the cold body isothermally at the lowest temperature, as in the Carnot cycle. This completes the third part of the cycle and constitutes the period of "**exhaust.**"

So far the Carnot cycle has been followed without variation.

(g) **Adiabatic Compression** (da). To complete the cycle, adiabatic compression should begin at the point d, so selected that when the piston reaches the end of the stroke, *all* of the working substance will be returned to the initial condition. But since a part of the steam has been reduced to water in the condenser and is no longer in the cylinder, it appears that the cycle cannot be completed by any process entirely within the cylinder itself.

It is possible, however, in theory at least, to complete the Carnot cycle by using a combined vapor-and-water pump, which, when the point d is reached, will receive all the water of condensation from the condenser and the vapor remaining in the cylinder at the point d, and by compression complete the condensation of the vapor, and bring the whole charge back to the initial condition by a process that is strictly adiabatic. But while this is conceivable it would be very difficult to carry out without introducing practical evils which would more than

counterbalance the thermodynamic advantage. In practice this last operation would be omitted, and, instead, the steam would either be expelled from the cylinder and condensed in a "condenser" or else exhausted into the atmosphere. The water of condensation, or an equivalent amount of "make-up water," is then pumped to the boiler, where the heat is added to bring the temperature *gradually* back to the initial value, which is not in accordance with the requirements of the Carnot cycle.

(h) It is true that in the actual steam engine compression is employed, but this must not be confused with the adiabatic compression of the Carnot cycle. But little of the steam is involved in this operation, and it is used principally for the purpose of "cushioning" the reciprocating parts in order to make the engine operate quietly. It has little effect on the thermodynamic operation of the engine.

(i) Although the Carnot cycle is not ordinarily followed by the steam engine, it is often very useful to determine the efficiency and the work that would be done with this cycle, within the temperature range of the steam engine, in order to find the maximum output that could be theoretically attained by *any* engine, using any kind of working substance with the same temperature limits.

Previous discussion of this cycle (Section 92) showed that for saturated steam the Carnot Cycle Efficiency, Ef_c, is given by Eq. (170), that the heat available, ΔQ_1, can be computed by Eqs. (172) to (174), and that the work done is $\Delta E = \Delta Q_1 \times Ef_c$ from Eq. (171).

Since superheat is supplied in practice with gradually increasing temperature, the Carnot cycle is not a satisfactory standard for comparison for engines using superheated steam, and hence this case will not be considered.

(j) As one h.p.-hr. is equivalent to the expenditure of 2545 B.t.u., and as each pound of steam makes available ΔE B.t.u. for doing external work, the Rate (W) of Steam Consumption per h.p.-hr., with the Carnot cycle, is evidently

$$W = \frac{2545}{\Delta E} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (225a)$$

$$= \frac{2545}{\Delta Q_1 \times Ef_c} \quad . \quad . \quad . \quad . \quad . \quad (225b)$$

(k) In practice some steam engines "exhaust" to the atmosphere, with the temperature of heat rejection theoretically equal to 212° F., corresponding to an absolute pressure of 14.7 pounds per square inch; while other engines exhaust to a condenser maintaining a vacuum of about 26″ of mercury, the absolute pressure being a little less than 2 pounds per square inch and temperature about 125° F. The steam turbine, which is one form of steam engine, is often operated with a vacuum of about 28″ of mercury or a little less than one pound "back pressure," the temperature being about 100° F.

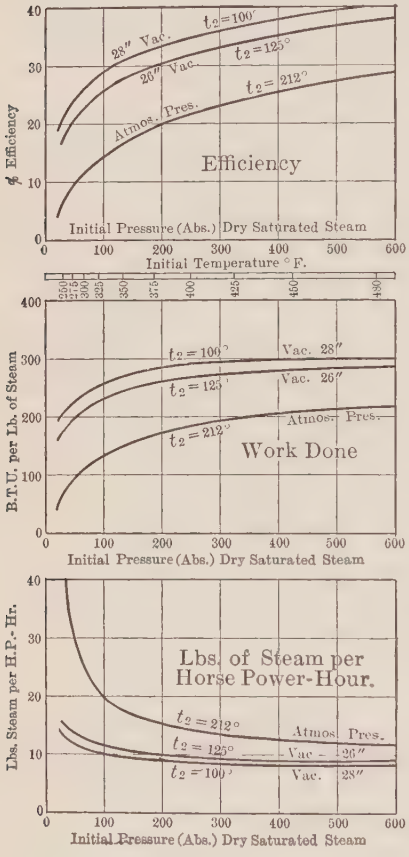


Fig. 73.

(l) Fig. 73 shows curves of efficiency, B.t.u. of work per pound of steam, and water rate, for the ideal engine operating on the Carnot cycle with steam initially dry saturated and with the three exhaust pressures mentioned above. A scale for saturation temperatures corresponding to the different initial pressures is also given.

(m) These curves show clearly that better results are obtained by increasing the initial temperature (or pressure) and by lowering the temperature (or pressure) of exhaust. A given temperature difference low on the temperature scale gives better efficiency than the same temperature difference at a higher range, as the denominator T_1 in Eq. (170) is lower. It is therefore theoretically advantageous to have T_2 as low as possible in any case.

109. The Regenerative Steam-Engine Cycle. (a) The Tϕ-diagram, Fig. 74, shows that if bc_1 is drawn parallel to the water line ad_1, the area abc_1d_1 will equal the area $abcd$ of the Carnot cycle. Thus, if steam is carried through the cycle abc_1d_1, and if heat is received only along the line ab, as in the Carnot cycle, the two cycles must have equal efficiencies.

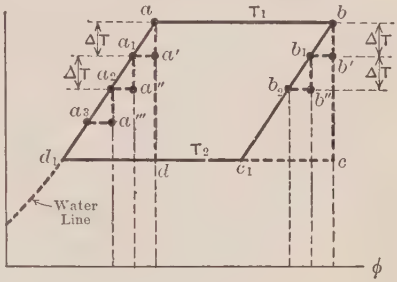


Fig. 74.

(b) The cycle abc_1d_1, called the Regenerative Cycle, can be obtained under ideal conditions in the following manner: While the steam is expanding an infinitesimal amount from b, with drop in temperature from T_1 to $(T_1 - \Delta T)$, let a sufficient quantity of the steam, or heat from the steam, be abstracted from the cylinder to cause the expansion to be along bb_1; and let this heat be used to raise the feed water from $(T_1 - \Delta T)$ to T_1, changing the water from condition a_1 to a. The heat abstracted from the cylinder during this process is shown by the area below b_1b; and that given to the water, by the equal area below a_1a. Similarly, while steam expands through another increment, b_1 to b_2, let sufficient heat be abstracted from the cylinder to raise the water from condition a_2 to a_1. Continue this process for each increment of expansion until the final temperature T_2 is reached. In this way the expansion bc_1 is made parallel to ad_1. Obviously, in each instance when heat is supplied to or abstracted from the working substance, the transfer is at a constant temperature (considering the ΔT as insignificantly small). Thus the surrender of heat by the steam and the reception of heat by the water correspond to the regenerative action in the Joule and the Stirling gas cycles. After the water has thus been brought to condition a, the boiler can supply the latent heat for vaporization at the constant temperature T_1; and when expansion has reached the point c_1, the heat is rejected at constant temperature T_2.

(c) As the reception of heat from the hot body and rejection of heat to the cold body are thus all isothermal and reversible processes, and as the temperature changes are equivalent to

adiabatic ones, this cycle is the equivalent of the Carnot cycle, and the equations of Section 92 and curves given in Section 108 for the latter can also be used for this Regenerative cycle.

(d) This Regenerative cycle has been used but little in practice. It is approximated in some engines built by Nordberg,* in which the steam is expanded in steps, by passing it successively through several cylinders. The theoretical expansion in the first cylinder corresponds to bb' in Fig. 74, but with a finite instead of an infinitesimal temperature drop; that in the second cylinder to b_1b''; and similarly for the expansions in each of the other cylinders. Heat represented by the area below b_1b' is abstracted from the steam from the first cylinder, (or from its steam jacket) and is used to raise the water from state a'' to a'; heat corresponding to the area below b_2b'', withdrawn from the second cylinder, raises the condition of the water from a''' to a''; and similarly from each of the other cylinders, heat is transferred to the water. Thus heat is abstracted by steps from the expanding steam and is used for progressively heating the feed water in small increments, each with but small rise in temperature. If these increments could be made infinitesimal, the heat additions would be isothermal and the Regenerative cycle would result. Nordberg used four steps only, but the remarkable results obtained with these engines when the heaters were in use, as compared with their performance without the heaters, seem to indicate that there may be considerable advantage to be gained by using the regenerative principle even with but few steps. One Nordberg engine attained 73.7 per cent of the efficiency of the Carnot cycle for the same temperature limits.

110. The Clausius Cycle. (a) The theoretical cycle, approximated by the ordinary steam engine, as was shown on pp. 195–197, is as follows: The working substance, starting as water at the boiling point, receives heat isothermally during the process of vaporization; it then expands adiabatically from the higher to the lower temperature; it is next condensed isothermally; and finally, after being returned to the boiler, is brought back to the initial state, not by adiabatic compression, but by the application of heat. This cycle will be recognized as the Clausius cycle (Sections 93 and 94), and not only is it the theoretical

* Transactions A. S. M. E., 1900, p. 181, and 1907, p. 705.

cycle of the steam engine, but also that of the steam turbine, as will be seen later; it therefore is of value not only in comparing the performances of steam engines with each other, but also in comparing engines with turbines.

(**b**) The Clausius cycle, with the *lower temperature taken as that of the exhaust steam*, was adopted in 1898 by the British Institution of Civil Engineers* as the standard of comparison for steam engines and turbines, but is called by them the "Rankine cycle," † it having been published simultaneously but independently by both Clausius and Rankine. The use of this temperature was subsequently adopted by the American Society of Mechanical Engineers in their "Power Test Code" of 1915. The temperature of the steam is measured in the exhaust pipe near the engine, or turbine, as the case may be.

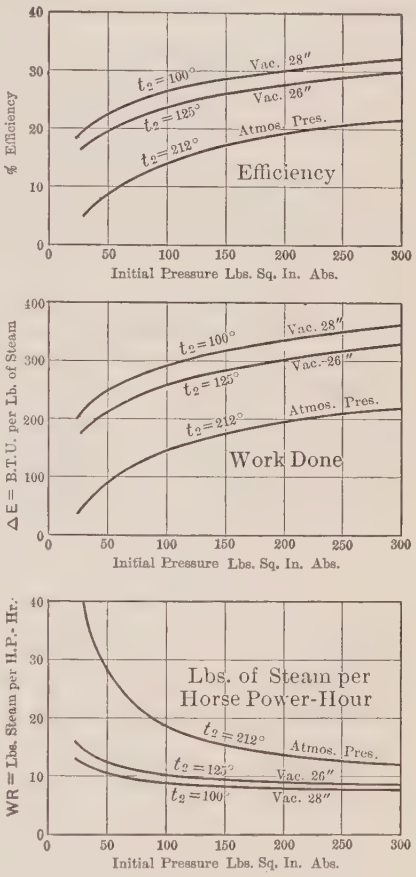


Fig. 75.—Clausius Cycle. Dry Saturated Steam.

(**c**) The work in B.t.u. per pound of steam (ΔE) can be computed for this cycle by using Eqs. (188) and (191), or it can be found directly and more conveniently from the Mollier chart, Plate II, in the Appendix. (For accurate results a larger chart should be used than is there given.)

The efficiency of the Clausius cycle can be computed from Eq. (192) or (193).

* Proceedings, 1898.

† Note that the name "Rankine cycle" is used in this book to designate a different cycle, i.e., the Clausius with incomplete expansion.

(d) Fig. 75 gives curves for the efficiencies, water rates, and work in B.t.u. for the three cases that were considered with the Carnot cycle, Section 108 (l), namely, with atmospheric exhaust pressures and with vacuums of 26″ and 28″ Hg.

(e) A comparison of these curves with those for the Carnot cycle (Fig. 73) shows lower efficiency in this case, as would be expected. The work per pound of steam is larger, however, which at first seems wrong, but which is explained by the fact that the heat supplied per pound from the source is much larger in the Clausius cycle than in the Carnot, being $(xr + q)_1 - q_2$ for the former as against x_1r_1 for the latter, when dry saturated steam is used. Thus with the Clausius cycle less weight of steam is used per h.p.-hr., but each pound receives more heat and this is used less efficiently than with the Carnot cycle.

(f) Comparing the Clausius cycle efficiencies when the steam is superheated with those when it is dry saturated (other conditions of operation remaining the same) shows that with superheat the efficiencies are so little higher that superheating would seem hardly worth while, when the additional expense of equipment and maintenance of superheating apparatus are considered. It will be seen later, however, that superheating may give beneficial results which are not in any way connected with the theoretical cycle, hence it is frequently used in steam-engine practice.

111. The Rankine Cycle. (a) In the reciprocating steam engine, instead of expanding to the point d_1, Fig. 76, it is the general practice to discontinue at some point d, and release the steam at constant volume, along the line de, as is done in what has been previously called the Rankine cycle (Sections 95 and 96). The "toe" of the diagram is thus cut off and the work represented by the area dd_1e is lost. The reason for sacrificing this work is twofold: (1) By reducing the maximum volume involved, that is, from V_{d1} to V_e, the size of the cylinder is proportionately decreased, and this results in material reduction of the size and cost of the engine. (2) There is a loss in power and effi-

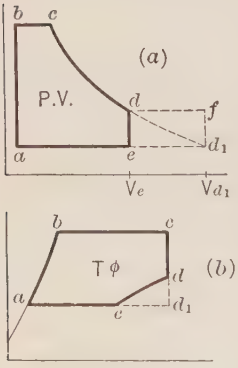


Fig. 76.

ciency if expansion is carried beyond the pressure that is just sufficient to overcome the frictional resistance of the engine; for in Fig. 76 (*a*), if expansion were completed, the additional work done by the steam is shown by the area edd_1; whereas, if *de* represents a pressure equal to the mean frictional resistance of the engine, the additional work in friction, with the greater piston movement *df*, would be given by the area ed_1fd, which is greater than the useful area edd_1, by the area dfd_1. The latter area represents the net loss accompanying an increase in expansion from *d* to d_1.

(b) In the steam turbine, which has very little mechanical friction, the expansion is continued down to the exhaust pressure, as in the Clausius cycle. That the reciprocating steam engine does not do the same is a fault chargeable against it and one that cannot be entirely remedied. Hence, as a standard of comparison for both the steam engine and the turbine, the Clausius cycle is preferable to the Rankine, and for that reason the latter cycle will not be further considered here.

112. Clearance and Compression. (a) In the theoretical cycles previously discussed it was considered that no steam space existed between the cylinder head and the piston at the beginning of the stroke, and hence that the initial volume of steam in the cylinder was zero. In practice, however, the piston must not touch the cylinder head, and the shortest distance between them is called the "*mechanical clearance*," with values from $\frac{1}{8}$ inch to $\frac{3}{4}$ inch or more. The cubical contents of this space, including the passages to the valves and all other spaces that must be filled with steam before the commencement of the stroke, is termed the "**clearance volume.**" It equals the initial volume on the PV-diagram, as shown by *cl* in Fig. 77. The percentage of "clearance volume" as compared with the piston displacement, or volume displaced by the piston per stroke, is from 2 to 15 per cent in practice.*

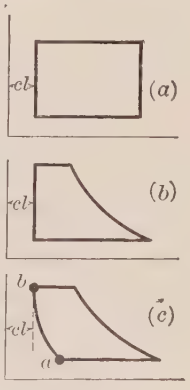


Fig. 77.

* The term "clearance" is used rather loosely as applying either to the linear or volumetric quantity, but the kind of clearance meant is usually apparent from the context.

(b) The clearance theoretically influences the amount of steam heat consumed per unit output of work. If the engine operates with a theoretical PV-diagram of rectangular form, Fig. 77 (*a*), the clearance space must be filled with steam each cycle, before the piston starts, and as nearly all of this steam is exhausted with that which performs the work, it theoretically represents a direct waste, the percentage of waste being equal to the percentage of clearance volume.

(c) If the cycle is that shown in Fig. 77 (*b*), a still greater proportion of the steam heat is wasted because of the clearance space; for in this case the clearance volume is greater in proportion to the total volume of steam admitted than in the preceding instance.

(d) If the cycle followed is that shown in Fig. 77 (*c*), in which there is adiabatic compression along *ab*, there is theoretically no loss due to the clearance; for at each stroke the weight of steam entrapped at *a* and compressed along *ab* can be considered, during expansion, as following back along *ba*, just as if this amount of steam were separated from the rest by a flexible diaphragm and were compressed and expanded adiabatically without being exhausted from the cylinder.

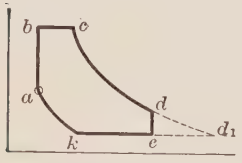


Fig. 78.

(e) If the compression is not carried to the initial pressure, but is along some line *ka* in Fig. 78, the case is intermediate between (c) and (d). If, in addition, the expansion is incomplete, terminating at some point *d* in Fig. 78, the clearance loss is theoretically a minimum when compression is carried to a point somewhat below *b*.*

(f) In the actual case there are certain influences to be discussed later, which modify the theoretical effect of compression. Whether or not compression in the actual engine improves the steam consumption is still a matter of discussion. The influence is so small that the effect is difficult to determine experimentally.

113. **Cushion Steam and Cylinder Feed.** (a) It is sometimes convenient to consider the weight of working substance present in an engine cylinder as composed of two parts, namely, that

* See Heck's "Steam Engine," Vol. I, p. 97.

entrapped during compression and that fed from the boiler during each cycle.

(b) The steam entrapped during compression is called the "**cushion steam.**" It is difficult to determine its quality throughout compression, but it is customary to assume it dry when compression begins. Since almost immediately after release the steam pressure drops to the back pressure, there is but little steam in the cylinder soon after the beginning of the back stroke. That part subsequently trapped in the clearance and compressed is subject to the higher temperature of the cylinder walls throughout nearly the whole of the return stroke, hence must be practically dry when compression begins.

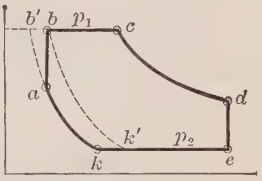


Fig. 79.

With this assumption, the weight of the cushion steam at k, Fig. 79, can be computed from

$$w_k = V_k \div \mathbf{V}_k, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (226)$$

in which V_k can be scaled from diagram and $\mathbf{V}_k$ is the specific volume, as given in the Steam Tables, for the pressure p_2 existing.

If the same weight of steam is raised to the initial pressure p_1 and is maintained dry and saturated, it will occupy the volume

$$V_b' = w_k \times \mathbf{V}_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (227)$$

where $\mathbf{V}_1$ is the specific volume at the initial pressure p_1. The volume V_b' is shown by the abscissa at the point b' in Fig. 79, in which $b'k$ is the saturation curve for a weight of steam equal to w_k.

Evidently when the valve opens to admit steam to the cylinder this weight w_k is already present, and it occupies the volume V_b' as soon as its pressure is raised to that of the entering steam.

(c) The steam that is supplied to the cylinder from the boiler at every cycle is called the "**cylinder feed**" (w_f). The cylinder feed may be determined by finding the weight of steam delivered to the engine in a given time and dividing this by the corresponding number of cycles.

If the boiler, without loss, furnishes to the engine all the steam it generates, the weight of steam supplied is equal to the weight of water fed to the boiler in the given time. If a surface condenser is used, w_f can be determined from the weight of the

steam condensed in the given time, provided there is no leakage in the condenser.

In Fig. 79 the volume of the cylinder feed is shown by the distance $b'c$. It would be represented by bc only in case the compression terminated at point b.

(d) The **total weight** (w) **of steam in the cylinder** during expansion is evidently made up of the cylinder feed (w_f) and the cushion steam (w_k); thus $w = w_f + w_k$. Its theoretical volume at the time of cut-off, if it is dry and saturated, is $V_c = w \times \mathbf{V}_c$, where $\mathbf{V}_c$ is the specific volume of the steam at the cut-off pressure.

114. Saturation and Quality Curves. (a) If the weight of steam (w) present in the cylinder is considered to be dry and saturated, it occupies at any time a volume $V_s = w\mathbf{V}$, where $\mathbf{V}$ is the specific volume for the pressure under consideration. By plotting on the PV-diagram the values of V_s for different pressures, a **Saturation Curve** is obtained. Such a curve is shown by cs in Fig. 80, and is of value in determining the quality of steam at different points during expansion (Section 70). For example, in Fig. 80, in which the expansion line cd is adiabatic, the quality of steam at any point C is $x = \dfrac{AC}{AS}$, and the "wetness factor" is $(1 - x) = \dfrac{CS}{AS}$.

(b) If qualities are determined for several points along the expansion line and are plotted as ordinates on the corresponding volumes, as in the upper part of Fig. 80, a **Curve of Qualities** is obtained, which shows how x varies during the expansion. The quality curve in this case shows the condensation that takes place in order to make heat available for doing external work during the adiabatic expansion. In the figure the steam is assumed to be dry and saturated at c, hence the saturation curve must pass through that point. Quality curves for any other kind of expansion line can be found in the same way; thus, if $c'y$ is the line, the quality at C' is $\dfrac{AC'}{AS}$, if the weight of material present is the same as before.

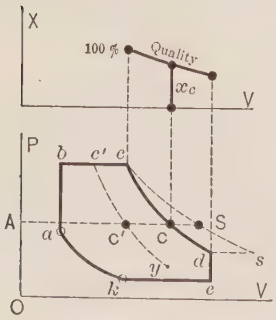


Fig. 80.

(c) Should the expansion line cross the saturation curve, as in Fig. 81, the quality ratio would be greater than 100 per cent, which would indicate that the steam becomes superheated during expansion. If the weight (w) of steam is known, the specific volume of the superheated material follows from $\mathbf{V} = \frac{V}{w}$, in which V is the volume scaled on the diagram. Then the absolute temperature of the steam may be computed by solving Tumlirz's Eq. (134). Thus $T = p\,(\mathbf{V} + 0.256) \div 0.5962$, in which p is the absolute pressure in pounds per square inch. By subtracting from T the absolute temperature of saturation at the pressure p the degrees of superheat D can be found.

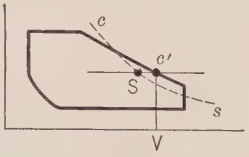


Fig. 81.

(d) On page 156 it was shown that under certain conditions the adiabatic expansion of wet steam can be represented quite accurately by PV^n = const., where n has different values which depend on the quality x at the beginning of expansion. The relation between n and x was given in Eq. (156), thus the initial quality can be determined if n is known. The value of n in any case can be found in several ways, but probably the most convenient method is to replot the expansion curve using logarithmic coördinates, in which case n is the slope of the expansion line (page 55). With the quality and volume known at the beginning of expansion, the corresponding weight of steam in the cylinder and the water rate can be determined readily.

CHAPTER XV.

ACTION OF STEAM IN REAL ENGINES.

115. Cylinder and Thermal Efficiencies of the Steam Engine. (General.) (a) The actual behavior of the steam in a cylinder is quite different from the theoretical, because of modifications of the ideal cycle, heat interchanges between the steam and the cylinder walls, and leakage of valves, piston, etc. The greater the perfection attained in the design, construction, and operation of the engine, cylinder, piston, valves, etc., the closer will the actual behavior approach the ideal. The measure of this perfection is given by the **indicated or cylinder efficiency** (IEf), Section 105 (e), which can be computed by Eq. (215), or by the following:

$$IEf = \frac{\textit{Actual B.t.u. of indicated work per lb. of steam}}{\textit{Theoretical B.t.u of work with Clausius cycle}} = \frac{H_i}{\Delta E} \quad (228a)$$

$$= \frac{\textit{B.t.u. equivalent of 1 i.h.p.-hr.}}{\textit{Heat available per i.h.p.-hr.}} = \frac{2545}{W_i \Delta E} \quad . \quad . \quad . \quad (228b)$$

$$= \frac{\textit{Theoretical lbs. of steam per h.p.-hr.}}{\textit{Actual lbs. of steam per i.h.p.-hr.}} = \frac{W}{W_i} \quad . \quad . \quad (228c)$$

In which

ΔE = B.t.u. work with Clausius cycle per lb. of steam.

H_i = Actual B.t.u. indicated work per lb. of steam $= \frac{2545}{W_i}$.

W = Pounds of steam theoretically needed per i.h.p.-hr. with Clausius cycle $= 2545 \div \Delta E$.

W_i = Pounds of steam actually used per i.h.p.-hr. as found by weighing the water used.

(b) The cylinder efficiencies of steam engines and turbines range from 40 per cent to 80 per cent, and in one exceptional case 88.2 per cent was attained. The reasons for the differences occurring between the real cycle and the Clausius, and for the losses which they represent, and the methods of reducing these losses, will be discussed in the succeeding sections.

If the same Clausius cycle is followed by two reciprocating steam engines, or by a reciprocating engine and a steam turbine, the ratio of the consumptions of steam, or heat, per i.h.p.-hr. of the two engines is equal to the inverse ratio of the cylinder efficiencies. If the theoretical cycles are not the same, such a comparison should not be made.

(c) It is at times necessary to predict the performance of a new engine, or turbine, when operating under certain definite conditions. In such cases the B.t.u. of work, ΔE, done by the Clausius cycle, per pound of steam, can be obtained from Eqs. (188) and (191), or from the Mollier or Ellenwood diagrams in the Appendix; then if the proper value of the Cylinder Efficiency (IEf) can be found, from data relating to similar engines operating under like conditions, the probable number of heat units that will be converted into work per pound of steam is, from Eq. (228a),

$$H_i = \Delta E \times IEf, \quad . \quad . \quad . \quad . \quad . \quad (229)$$

and, from Eq. (228b), the probable steam consumption per i.h.p.-hr. is

$$W_i = \frac{2545}{(\Delta E \times IEf)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (230)$$

(d) The **Thermal Efficiency on the i.h.p.** ($TIEf$), Section 105 (g), is the ratio of the indicated work to the heat supplied for doing this work; it is therefore a measure of combined efficiency of the cycle and of the cylinder with its appurtenances.

The engine cannot use heat that is of temperature lower than that of the exhaust steam T_2, and, theoretically at least, the heat remaining in the condensate can be returned to the boiler with the feed water; hence the heat of the liquid below this lower temperature is not chargeable against the engine.

Therefore,

$$TIEf = \frac{\textit{B.t.u. indicated work per lb. of steam}}{\textit{Heat supplied above } T_2 \textit{ per lb. of steam}} \quad . \quad (231a)$$

$$= \frac{H_i}{(\Delta Q_1 - q_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (231b)$$

$$= \frac{2545}{W_i\,(\Delta Q_1 - q_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (231c)$$

Where H_i = B.t.u. of actual indicated work per pound of steam

$$= 2545 \div W_i = \Delta E \times IEf \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (232)$$

$$\Delta Q_1 = (xr + q + \overline{C}_p D)_1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (233)$$

q_2 = heat of liquid above 32 degrees when at the temperature T_2 of the exhaust steam.

W_i = Pounds of steam actually used per i.h.p.-hr. as found by weighing the water used.

The value of $TIEf$ varies with the kind of engine and conditions of operation, and ranges in practice from 5 per cent to 25.05 per cent, this latter value being the maximum yet recorded.

In the case of a new engine, the probable performance may be computed, if the value of $TIEf$ for similar engines and conditions are known, by using the following equations, derived from Eq. (216):

$$W_i = 2545 \div \{(\Delta Q_1 - q_2) \times TIEf\} \quad . \quad . \quad (234)$$

$$H_i = (\Delta Q_1 - q_2) \times TIEf. \quad . \quad . \quad . \quad . \quad . \quad (235)$$

(e) The **Mechanical Efficiency** (MEf) of the steam engine mechanism varies from 85 to 97 per cent.

(f) The measure of the combination of the efficiencies of the cycle, cylinder, and engine mechanism is the **Thermal Efficiency on the d.h.p.** ($TDEf$), Section 105 (**h**). The $TDEf$ can be computed from Eqs. (219) and (220). Since it is the ratio of the B.t.u. of work delivered, to the heat supplied in doing that work,

$$TDEf = \frac{2545}{W_d\,(\Delta Q_1 - q_2)}, \quad . \quad . \quad . \quad . \quad (236)$$

in which W_d is the weight of working substance supplied to the engine per d.h.p.-hr., and ΔQ_1 is the heat per pound as given by Eq. (233), and q_2 is the heat of the liquid at exhaust temperature T_2.

The $TDEf$ is from 85 to 97 per cent of the $TIEf$, the ratio being equal to the mechanical efficiency.

The probable performance of an engine on the basis of delivered power may be estimated by using the following equations,

$$W_d = 2545 \div \{(\Delta Q_1 - q_2) \times TDEf\} = W_i \div MEf \quad (237)$$

and

$$H_d = TDEf\,(\Delta Q_1 - q_2), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (238)$$

where H_d is the B.t.u. of work delivered per pound of steam.

(g) The measure of the performance of the engine *as a whole* as compared with the ideal engine following the Clausius cycle is given by the **Overall Efficiency** (*OEf*). This is the combined efficiency of cylinder and mechanism, hence $OEf = IEf \times MEf$ as in Eq. (221). Since it is the ratio of work actually delivered to that which would be delivered by the ideal engine,

$$OEf = \frac{2545}{W_d \times \Delta E}, \quad . \quad . \quad . \quad . \quad . \quad (239)$$

in which W_d is the weight of steam actually used per d.h.p.-hr., and ΔE is work theoretically obtainable per pound of steam following the Clausius cycle. ΔE can be computed by using Eqs. (188) to (191).

The overall efficiency of steam engines is from 35 to 78 per cent.

The probable performance of the steam engine, based on the delivered output, can be estimated from

$$W_d = 2545 \div (\Delta E \times OEf) \quad . \quad . \quad . \quad . \quad . \quad (240)$$

and

$$H_d = \Delta E \times OEf. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (241)$$

116. Actual Behavior of Steam in an Engine Cylinder. (a) The materials with which the steam comes in contact in the engines previously considered have been assumed to be perfect nonconductors of heat; but materials actually used are good heat conductors, and this modifies the behavior of steam in real engines.

(b) The steam on its way from boiler to engine gives up to the pipe a part of its heat, which is lost by conduction, radiation, and convection. Because of this, the steam arrives at the engine with quality or superheat reduced. There is another loss due to the drop in pressure necessary to cause the steam to flow from the boiler to the engine against resistances due to friction of the pipe and inertia of the steam itself.*

(c) At the engine the steam on its way to the cylinder passes through a "Throttle Valve." If this valve is only partly open, the steam is **"Throttled"** or **"Wire-drawn,"** with an accompanying drop in pressure. These changes take place with such

*This kind of loss will be considered later in the chapter on flow of steam through pipes.

great rapidity and within such small space that little heat loss to the outside can occur, but as there is an increase in the velocity of the steam there is a small amount of heat expended in imparting kinetic energy. This, however, returns as heat when the velocity is reduced in the engine. The heat lost while the pressure is decreased is so small as to be negligible, hence the process may be considered one in which the *total associated heat remains constant.* The throttling increases the quality (or superheat), the value of which can readily be found by following along the proper constant-heat line either on the Mollier diagram or on the Tϕ-diagram, from the point for the initial condition to that for the lower pressure (or temperature).

A further wire-drawing takes place, with a similar effect on the condition of the steam, while the steam passes the more or less restricted opening of the admission valve on its way to the cylinder.

(d) When the steam enters the clearance space in the cylinder, it comes in contact with surfaces that were cooled during the period of exhaust of the preceding cycle. In the resulting interchange of heat, the temperature of the clearance walls is raised, and ordinarily from 10 to 40 per cent of the steam is condensed. The corresponding decrease in the quality of the steam is shown by the inclination of the quality curve x_1b_1 in Fig. 82.

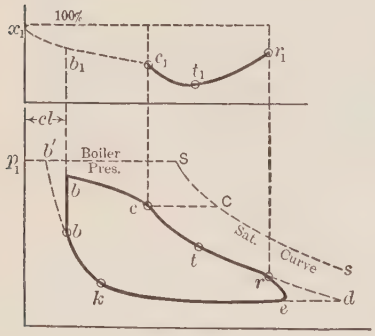


Fig. 82.

As the piston recedes during admission (bc), an increasing amount of the cylinder wall is exposed to the entering steam and further condensation takes place (as shown by the slope of b_1c_1) until, at the point of cut-off, from 20 to 50 per cent of the steam has usually been condensed. Next to the heat loss inherent in the theoretical cycle, this **Initial Condensation,** as it is called, causes the largest loss that ordinarily occurs in the steam engine and is therefore the one most desirable to minimize.

(e) During admission the pressure is decreased by the wire-drawing of the steam while passing the admission valve and

while flowing through the passages to the cylinder. This causes small loss of heat but decreases availability. It improves the quality or superheat of the entering steam somewhat and may reduce the initial condensation a very slight amount. The decrease of pressure by wire-drawing and the effect of condensation during admission are shown by the downward slope of the admission line, *bc*.

(**f**) After cut-off (*c*), the condensation continues, until expansion has reached a point (*t*) where the temperature of the steam equals the mean temperature of the exposed cylinder walls. The accompanying change in quality is shown by the curve c_1t_1. With further expansion the average quality of steam increases; steam is still condensed on the surfaces newly uncovered by the continued motion of the piston, but the heat thus absorbed by the cooler portion of the cylinder wall is less than that given up, to evaporate moisture, by the rest of the wall which is at relatively higher temperature. The increase in quality is shown by t_1r_1. Of course, the condensation during expansion is not all due to the influence of the cylinder walls, for heat must be used in performing the external work, as was shown in Section 114 (**b**).

The quality curve $c_1t_1r_1$, for the period of expansion, is readily obtainable after the saturation curve has been drawn for the total weight of mixture in the cylinder; but that part of the quality curve which relates to admission ($x_1b_1c_1$) is indeterminate and is therefore shown dotted in the figure.

(**g**) Theoretically, expansion should be continued to the point *d* on the back pressure line; therefore, in the actual case there is a loss due to **incomplete expansion,*** measured by the area *rde*.

During release (*re*) the steam is dried to some extent by the heat that is given up by the cylinder wall to the steam, which is now at a low temperature, and also by the heat released from the steam itself while decreasing in pressure.

(**h**) During exhaust (*ek*) the confining walls are cooled by the outflowing steam, and by the evaporation of the film of moisture on the walls. The greater the amount of moisture, within limits, the cooler will the walls become and the greater will be the amount of steam condensed during admission in the

* See Section 111.

next stroke. The exhaust pressure, or **back pressure,** is somewhat higher than the theoretical because of the resistance to steam flow offered by the valve opening and passages, by the inertia of the steam itself, and because of the evaporation of moisture.

(i) During compression the quality of the steam is indeterminate. It is usually assumed, however, that the steam is dry at the beginning of compression, and this is accurate enough for most practical purposes because of the small weight of steam involved. During the first part of compression the steam will probably be slightly superheated owing to the reception of heat from the hotter walls of the cylinder. If the compression is high, the temperature from compression may rise above that of the cylinder walls, in which case condensation will follow. Further compression of the now saturated steam will be at constant pressure, and on the PV-diagram the line becomes horizontal, thus forming the " hook " as in Fig. 83.*

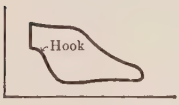


Fig. 83.

(j) The cycle is further influenced by the **leakage** of steam past the valves and around the piston, which would modify the actual diagram.

(k) The loss of heat from the cylinder walls by **radiation, conduction,** and **convection** lowers the mean temperature of the walls and adds slightly to the condensation.

117. Diagrammatic Representation of the Heat Interchange in the Cylinder. (a) In the **PV-diagram,** Fig. 84 (*a*), the point

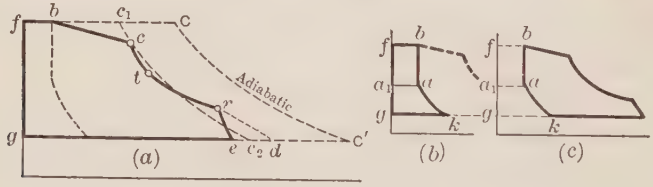


Fig. 84.

C represents the total charge of steam and water in the cylinder, considered raised to the initial pressure and in a dry saturated condition. *CC'* is an adiabatic expansion line drawn through

* A somewhat similar hook occurs when there is leakage past the piston or valve.

this point, and $fCC'g$ is the Clausius diagram which this charge should theoretically give. The actual diagram (omitting compression and clearance) is *fbcreg*. If through c the adiabatic c_1c_2 is drawn, then fc_1c_2g is the Clausius diagram for the vapor actually present at cut-off. Thus the loss of area from initial condensation is seen to be $c_1CC'c_2$, and that due to wire drawing is bcc_1. That the expansion line from c to t has greater slope than cc_2 shows that the condensation takes place more rapidly than it would with adiabatic expansion. At t the temperature of the steam is the same as that of the cylinder walls, and from t to r reëvaporation takes place, as shown by the expansion line being more nearly horizontal than the adiabatic.

The loss due to early release is shown by the area *rde*, in which *rd* is an adiabatic line.

(**b**) The diagram for the compression of the cushion steam is shown separately in Fig. 84 (*b*) by $gkaa_1$; and above it is the area a_1abf for admission at constant volume. By subtracting Fig. 84 (*b*) from Fig. 84 (*a*) the net diagram is obtained, as in Fig. 84 (*c*).

(**c**) The use of a **Tϕ-Diagram** to represent the heat interchanges occurring in the cylinder is more or less conventional and is only partly correct from the theoretical standpoint. In

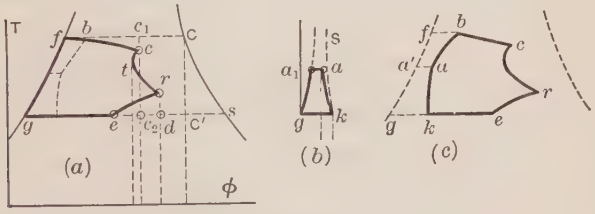


Fig. 85.

Fig. 85, which is lettered to correspond with Fig. 84, $fCC'g$ is the Clausius diagram for the total weight of charge used per cycle; *fbcreg* conventionally represents the actual diagram, neglecting clearance and compression; and fc_1c_2g is the Clausius cycle for the vapor actually present at the time of cut-off. The loss of heat to the cylinder wall due to initial condensation is given by area below c_1C down to the ϕ-axis; but as that part below c_2C' could not be utilized, the net loss of area from this cause is $c_1CC'c_2$. The wire-drawing loss is represented

approximately* by the area bcc_1. That heat is lost to the cylinder wall after cut-off is shown by the sloping of the expansion line ct to the left of the adiabatic cc_2, which indicates that the quality decreases more rapidly than it would with adiabatic expansion. At t condensation ceases; and from t to r reëvaporation takes place, accompanied by an increase in the quality. The loss due to early release is represented approximately by the area erd.

(d) The Tϕ-diagram for compression of the small weight of cushion steam is shown in Fig. 85 (b) by the negative area kaa_1g drawn to the same scale as before, except that the width has been reduced in proportion to the weight of working substance involved. Here it is assumed that the steam is dry at the beginning of compression, in which case the saturation curve S would pass through the point k. The heat lost to the cylinder walls during compression is shown by the area under ka, and the qualities during compression can be readily determined in the usual manner on such diagrams.

In Fig. 85 (c) the intercepts between $a'g$ and ak are the same as those between the similar lines in Fig. 85 (b); ab is the line for the constant volume during admission (corresponding to ab in Fig. 84), and $fbcreg$ is the same as in Fig. 85 (a). The area $abcrek$ evidently approximately represents the work done during the actual cycle.

(e) Obviously, cr is the only part of the diagram that shows the true behavior of *all* the steam used per cycle; for, during the other parts of the cycle, only a part of the steam is within the cylinder. As the Tϕ-diagram is ordinarily used in connection with the actual steam engine, it is assumed that if x parts of dry steam are present, the associated heat is the same as that of all the steam when at quality x, and the state point is located on the diagram accordingly. This is fallacious, since $x\,(r + q)$ is not the equivalent of $(xr + q)$. Thus, the Tϕ-diagram is correct only for the expansion process (the weight of material being constant); it is erroneous to use it quantitatively for the other processes, but it shows in a general way what interchanges occur in these other cases. As the Tϕ-diagram is ordinarily interpreted, it would be said that while tracing the line from b to c the quality would be increasing, but in this case it is the

* The reason this is approximate will appear later.

volume of the vapor that is increasing in the cylinder, while at the same time its quality is usually decreasing. Again, while *ck* is being drawn, the volume of the steam in the cylinder is decreasing, and the quality is increasing.

(f) In the foregoing discussion the upper and lower temperatures were taken as those occurring in the cylinder itself. If isothermals corresponding to the boiler and condenser temperatures are drawn on the diagram, the added areas would show the losses between the boiler and cylinder and between the cylinder and condenser.

118. Derivation of a Tϕ-Diagram from a PV-Diagram. (a) **First Method.** On the PV-diagram draw the saturation curve for the total weight of mixture (w) involved; take numerous points around the diagram; and for each get the temperature (T), and the ratio (X) of the actual volume to that of dry saturated steam as given by the saturation line. Note that during expansion X will be the quality, while during other parts of the cycle it is simply a ratio of volumes.

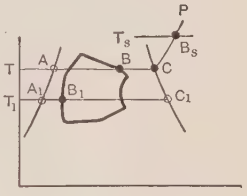


Fig. 86.

Prepare a Tϕ-chart as in Fig. 86, by drawing the water and saturation curves. This may be done conveniently by using absolute temperatures and entropies of water and vapor given in the Steam Tables, the entropies being multiplied by w before plotting. For each value of T draw the isothermal line TC. Then the distance AC is the entropy of vaporization. Locate the point B on AC in such position as to make $\frac{AB}{AC}$ equal to the value of X obtained from PV-diagram. The locus of points B thus found will be the desired diagram.

(b) The foregoing applies only to saturated steam. If X should be greater than 100 per cent, the temperature (T_s) of the superheated steam must first be found, and this may be done by using Tumlirz's formula in the manner explained in Section 114 (c); then the corresponding point B_s (Fig. 86) must be located in the region of superheat on the pressure line P at the temperature elevation T_s.

(c) If a Tϕ-chart for one pound instead of for (w) pounds is constructed like Plate I in the Appendix, it may be used for

the derivation of a Tϕ-diagram by plotting corresponding values of T and X directly; and it can be used regardless of the weight of steam involved, thus avoiding the construction of a new chart for each case. It must be remembered, however, that the areas on such diagrams represent the heat for only one pound of steam.

(d) **Second Method (Graphical).** In this method it is first necessary to prepare a **Boulvin Chart*** such as is shown in Fig. 87, in which there are four quadrants with related co-

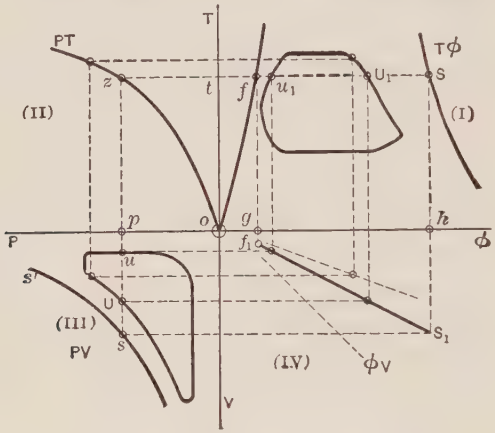


Fig. 87.

ordinates. The first quadrant (I) is for temperature-entropy (Tϕ) relationships; the second (II) for temperature-pressure (TP); the third (III) for pressure-volume (PV); and the fourth (IV) for entropy-volume (ϕV).

(e) In the PV-quadrant let the saturation curve ss', for the weight of steam w, be drawn with any convenient scales for the pressures and volumes. Then, in the PT-quadrant, plot a curve showing the pressure-temperature relation for saturated steam, using the same pressure scale as before and any suitable one for absolute temperatures. Next, in the Tϕ-quadrant, using the same temperature scale and any convenient one for entropy, construct the curves for water and for saturated steam. Then for any pressure p, the chart shows that the volume of the saturated steam is ps; that the temperature is $pz = ot$;

* "Entropy Diagram," by J. Boulvin, published by Spon and Chamberlain.

that the entropy of the water is tf; and that the entropy of vaporization is fS.

(f) While steam is being generated, the entropy of vaporization increases uniformly with the volume of vapor formed. For the particular pressure, p, under consideration, this relation may be shown in the remaining fourth quadrant (IV) by the straight ϕV-curve f_1S_1. This curve is obtained by projecting downward from the points f and S on the Tϕ-diagram and by making gf_1 = the volume occupied by the water $= w \times 0.017$,* and $hS_1 = ps$ = (the volume of w pounds of dry saturated vapor). To complete the chart, similar ϕV lines must be drawn for each of the other pressures used.

(g) On this chart the actual PV-diagram can now be drawn in the PV-quadrant (III) and from it the corresponding Tϕ-diagram can be obtained by simple projection. For example, starting with the points u and U (at the pressure p) on the PV-diagram, project horizontally to the ϕV-curve, f_1S_1, for that pressure, and thence upward to intersect the corresponding isothermal line at u_1 and U_1. The points thus found are on the Tϕ-diagram desired and other points can be located in a similar manner. By passing curves through the points, the complete diagram is obtained.

(h) With superheated steam, this construction does not apply, and in this case the procedure would be that outlined in (b) of this section.

(i) If a **Chart for One Pound of Steam** is constructed, it may be used for any case regardless of the weight (w) of steam involved. Then, however, the volumes to be used on the chart are $\left(\frac{1}{w}\right)$th of the actual volumes occupied by the steam in the cylinder; and the areas represent the work, or heat, for only *one* pound of steam.

119. Hirn's Analysis. (a) If certain data, which can readily be obtained during an engine test and from the indicator diagram, are available, the numerical values of the heat interchanges between the cylinder walls and the steam can be calculated by a method originated by Hirn (in 1876) and formulated later by Dwelshauvers-Dery.

With such information before him, the engineer can analyze the distribution and extent of the losses in each case, and, by comparing these results with those obtained with the best engines,

* This is too small to be scalable, but is shown exaggerated in Fig. 87.

he can determine wherein improvements can be made in the engine he is considering.

(b) With the weight of steam per cycle, and the pressure and quality of steam known at any two points (1 and 2) in the cycle, the associated heat (H_1 and H_2) present in the steam at those points can be computed. Then $(H_1 - H_2)$, if positive, is the heat surrendered by the steam between the two points; and if negative, it is the heat the steam receives. The B.t.u. work (A) actually done between points 1 and 2 of the cycle is shown on the indicator diagram by the area below the cycle line between those points and extending to the line of absolute zero pressure. Of course, if all the heat that is available is converted into work, A will equal $(H_1 - H_2)$. In the actual case, however, there is some heat interchange between the cylinder walls and the steam. Thus, if A is less than $(H_1 - H_2)$, the steam has lost heat to the cylinder walls equal to the difference; and if A is greater, heat has been surrendered by the walls to the steam and has been converted into work.

(c) The data needed for Hirn's analysis are:

(1) The weight (w_f) of "cylinder feed" per cycle and its quality (x_f) as it enters the cylinder, as determined by test of engine. This gives means of computing the heat H_f supplied by the entering steam.

(2) The weight of "cushion steam" (w_k) per cycle and its quality (x_k) at the beginning of compression.

(3) An average indicator card, with PV-axes, saturation curve, and quality curve, as in Fig. 88 (a) and (b).

(4) The B.t.u. equivalent of work per cycle as determined from the areas A_a, A_c, A_r, A_e, and A_k on the diagram, Fig. 88 (c) and (d).

(5) The heat (K_1) in the water of condensation, and the heat (K_2) carried away by the condensing water, supposing a surface condenser is used.

The leakage must be practically zero and is considered such in this analysis. Account must also be taken of the fact that during the reception of steam at constant pressure, the APu quantity is abstracted; thus throughout expansion and compression the heat *in* the steam is $(x\rho + q)$ instead of $(xr + q)$ for saturated steam, and is $(\lambda + \overline{C}_p D - APu_s)$,* instead of $\lambda + \overline{C}_p D$,

*Here $u_s = (V_s - 0.017)$ = the increase in volume accompanying the formation of superheated steam from one pound of water, i.e., it is the increase of volume during vaporization and during superheating.

for superheated steam. This also is true of the steam contained in the cylinder at each point in the cycle, because at some time before the point is reached the piston has moved out against resistance to make available the necessary volume and the APu quantity has thus been utilized.

(**d**) During the **first part of the cycle,** heat (H_f) is supplied by the entering steam (cylinder feed) and this is added to the heat (H_a) in the cushion steam at the end of compression.

The heat associated with (" in ") the entering " cylinder feed " is, in the case of saturated steam,

$$H_f = w_f (xr + q)_f, \quad . \quad . \quad . \quad . \quad . \quad (242)$$

and for superheated steam is

$$H_f = w_f (\lambda + \overline{C}_p D)_f. \quad . \quad . \quad . \quad . \quad . \quad (243)$$

At k (Fig. 88) the steam is assumed to have 100 per cent quality, as explained in Section 113 (**b**); hence the weight of the cushion steam is $w_k = \frac{V_k}{\mathbf{V}_k}$, where V_k is the absolute volume at k and $\mathbf{V}_k$ is the specific volume for the pressure at that point.

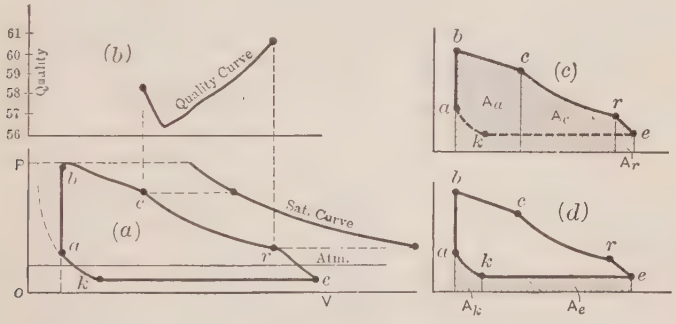


Fig. 88.

The weight of that part of the cylinder content that is in the form of vapor at a is similarly $w_a = \frac{V_a}{\mathbf{V}_a}$. Hence the quality at a is $x_a = \frac{w_a}{w_k}$ and the heat in the steam at this point is

$$H_a = w_k (x\rho + q)_a. \quad . \quad . \quad . \quad . \quad . \quad (244)$$

Then at the end of admission (point c) the heat in the steam is

$$H_c = (w_k + w_f) (x\rho + q)_c, \quad . \quad . \quad . \quad . \quad (245)$$

in which x_c is found from the quality curve (Fig. 88 (b)).

The work in B.t.u. per cycle actually done on the piston during admission is A_a, as determined from diagram Fig. 88 (*c*); and the heat given up by the steam is $(H_f + H_a) - H_c$; hence the heat interchange during admission is

$$L_a = (H_a + H_f) - H_c - A_a. \quad . \quad . \quad (246)$$

A positive result indicates that heat is lost to the cylinder and a negative one shows that the steam has received heat from the cylinder. The same will be true for the other equations for heat interchange that follow.

The proportion of heat that is interchanged is $L_a \div (H_a + H_f - H_c)$, and this is a close measure of the proportion of steam that is condensed, that is, it is a measure of the " initial condensation."

(e) At the beginning of **expansion** the heat in steam (at *c*) is H_c from Eq. (245).

The heat in steam at *r* is

$$H_r = (w_k + w_f)(x\rho + q)_r, \quad . \quad . \quad . \quad . \quad (247)$$

in which x_r is obtained from the quality curve in Fig. 88 (*b*).

The work in B.t.u. actually done during expansion is determined from area A_c on the diagram.

Then the net heat transfer from steam to cylinder wall during expansion is

$$L_c = (H_c - H_r) - A_c. \quad . \quad . \quad . \quad . \quad (248)$$

A negative result indicates that the steam receives heat from the cylinder wall.

Note that the loss between cut-off and any other point on the expansion line can be computed in a similar manner; thus it is possible to determine the interchanges between cut-off and all points throughout expansion.

(f) If the **exhaust** steam is condensed in a surface condenser, the condensate (of temperature t_e) per cycle will contain heat above 32 degrees,

$$K_1 = w_f (t_e - 32°), \text{ or more accurately } = w_f q_e; \quad . \quad (249)$$

and the condensing water (of weight w_x per cycle, with initial temperature t_0 and discharge temperature t_d) will take away heat,

$$K_2 = w_x (t_d - t_0), \text{ or more accurately, } = w_x (q_d - q_0). \quad (250)$$

Between r and k, the steam for a while does work, as shown by A_r on the indicator diagram; afterwards work is done *upon* it, as shown by A_e. At r the heat in the steam is H_r (from Eq. (247)), and at k there is left in the steam $H_k = w_k(\rho + q)_k$, since x_k is taken as unity.

Hence the heat interchange during exhaust is

$$L_e = (H_r - H_k) - (K_1 + K_2) - (A_r - A_e), \quad . \quad (251)$$

in which positive and negative results have the same meanings as explained in connection with Eq. (246).

(g) If the steam is exhausted to the atmosphere, the heat discharged is indeterminate. An approximation can be made if the mean quality of the exhaust steam is known, but it cannot be computed accurately because the weight, pressure, and quality of steam are variable throughout the exhaust period.

(h) During **compression** the change of associated heat is

$$(H_k - H_a),$$

in which

$$H_k = w_k(\rho + q)_k. \quad . \quad . \quad . \quad . \quad . \quad (252)$$

and

$$H_a = w_k(x\rho + q)_a \quad . \quad . \quad . \quad . \quad . \quad (253)$$

The work actually done upon the steam per cycle is shown by A_k. Hence the heat interchange during compression is

$$L_k = H_k - H_a + A_k, \quad . \quad . \quad . \quad . \quad . \quad (254)$$

in which the sign of L_k has the same meaning as before.

(i) Ideally, the heat given to the cylinder walls should equal that given up by them to the steam. Actually, in the case of an ordinary engine, the heat given up is less than that received, and this is because of conduction and radiation. Evidently, the **conduction and radiation loss** in B.t.u. per cycle is

$$R = L_a + L_c + L_r + L_k. \quad . \quad . \quad . \quad (255)$$

(j) If the steam is initially superheated, the analysis would be carried through in a manner similar to that just given. If the engine is steam-jacketed (Section 129), or is a compound engine with reheating receivers (Section 130), account must be taken of the heat furnished by the jacket steam.

120. Experimental Determination of the Actual Performance of Steam Engines. (a) The **indicated horse power** of an engine can be determined from the indicator diagram, if the diameter

of cylinder, length of stroke, and r.p.m. of the engine are known. The delivered horse power can be measured by a Prony brake or other form of dynamometer and in other ways which need not be considered here. The **mechanical efficiency** of the engine and power lost in **engine friction** can then be calculated. The total amount of **steam used per hour** may be determined by weighing the water pumped to the boiler which supplies the engine, making proper allowance for loss or withdrawal of working substance between the pump and the engine; or it may be found by weighing the condensate, if a surface condenser is used, and making correction for leakage. The weight of steam used per h.p.-hr., or **Water Rate,** can then be obtained as in Section 106.

(**b**) If these measurements are made for a range of loads on the engine, the resulting data can be used in plotting curves of Total Consumption of Steam (TC) and of Rate of Consumption (R), as in Fig. 71. These curves show the performance of the engine under all conditions of loading, and determine the power output at which the engine operates most economically. The curve of total consumption is usually nearly straight. If the abscissas are d.h.p.-hr., the Y-intercept shows the steam used in overcoming the friction of the engine alone.

(**c**) If two engines receive steam of the same pressure and quality, their relative performance is shown by comparing their Water Rates. In other cases the only true measure of economy is on the basis of *heat* used per unit of power, and in order to determine the heat in the steam as it enters the engine the quality, or superheat, must be known.

The quality of the steam can be determined by using instruments which will be described in the next section.

121. Steam Calorimeters.* (**a**) The apparatus used to determine the quality of steam is called a "steam calorimeter." There are several kinds of calorimeters, which will be considered very briefly.

The Barrel Calorimeter.

(**b**) If into a barrel containing water, of known weight (W) and temperature (t_1), a sample of the steam is piped, and con-

* For more detailed discussion see Carpenter and Diederichs' "Experimental Engineering," published by John Wiley & Sons.

densed, and if the increase (w) in weight of water and the resulting temperature (t_2) are measured simultaneously, there are sufficient data for determining the quality of the sample of steam, provided the steam pressure is known.

The heat given up by the steam is

$$\Delta Q_1 = w\{xr + q - (t_2 - 32°)\}$$

and that received by water is

$$\Delta Q_2 = W(t_2 - t_1),$$

assuming C_p of the liquid as unity. Evidently $\Delta Q_1 = \Delta Q_2$, from which the quality is found to be

$$x = \frac{W(t_2 - t_1) - w(q + 32 - t_2)}{wr}. \quad (256)$$

Correction should also be made for conduction and radiation losses in accurate work.

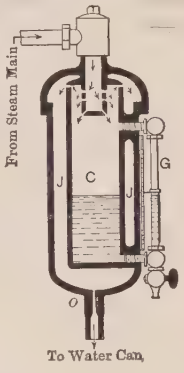


Fig. 89.

The Separating Calorimeter.

(c) In using the Separating Calorimeter (Fig. 89), a sample of steam is first led to the separating chamber C, where the moisture is thrown out and collected (the amount w being shown by the gauge glass G), then the resulting dry steam passes into the jacket J and out through the orifice O to a can of water in which it is condensed and its weight W determined. Using simultaneous values of w and W, the quality evidently is

$$x = \frac{W}{w + W}. \quad \ldots\ldots\ (257)$$

The Throttling Calorimeter.

(d) In this case a sample of wet steam is passed through the device shown in Fig. 90, and is superheated by being throttled through the valve V while expanding into the cup C, where the pressure is low.

This pressure is usually nearly atmospheric when high-pressure steam is being sampled. If the sample of steam is at pressure near or below that of the atmosphere, the cup may be connected with a condenser to obtain a sufficiently low pressure therein.

The temperature t_s of the superheated steam in this cup C is

measured by the thermometer T; and the degrees of superheat D are found by subtracting from t_s the saturation temperature t_2 corresponding to the cup pressure shown by the manometer.

The expansion through the valve causes the jet of steam to acquire a high velocity at that point, hence some of the associated heat is converted into kinetic energy. In the cup, the velocity of steam is reduced and this kinetic energy is reconverted into associated heat. If the velocity in the cup, where the temperature t_s is measured, is the same as that in the main steam pipe (which is usually approximately true), and if there are no radia-

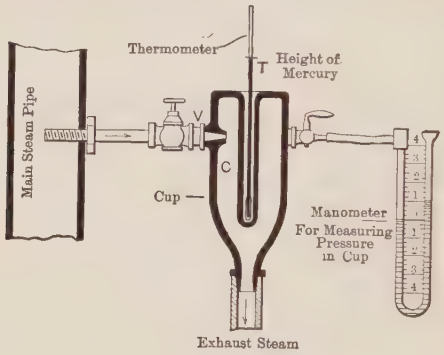


Fig. 90.

tion or conduction losses (and these are usually almost negligible), the associated heat is the same before and after the steam passes the expansion valve.

Before throttling, the amount of heat per pound is

$$\Delta Q_1 = x_1 r_1 + q_1;$$

afterward, it is

$$\Delta Q_2 = \lambda_2 + \overline{C}_p (D) = \lambda_2 + 0.48^* (t_s - t_2).$$

Then since

$$\Delta Q_1 = \Delta Q_2,$$

the quality is found to be

$$x_1 = \frac{\lambda_2 + 0.48^* (t_s - t_2) - q_1}{r_1} \quad . \quad . \quad . \quad . \quad . \quad (258)$$

* For cup pressures other than atmospheric substitute the proper value of $\bar{C}_p$ for 0.48.

The Electric Calorimeter.

(e) In using this calorimeter the sample of wet steam is dried by letting it flow over coils of wire which are heated by an electric current, the energy input being measured by a wattmeter. The watts are gradually increased until a value E is reached at which the thermometer in the calorimeter outlet starts to rise, which is supposed to show that all moisture has been dried by the heat from the coils, by expending an amount of heat corresponding to E. If the quantity of mixture (w) flowing through the calorimeter in a given time is weighed, or otherwise determined, the heat (h) added to dry one pound of the steam can be computed from the electrical input. Then

$$x = \frac{r - h}{r} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (259)$$

The Degrees of Superheat.

(f) This is determined by subtracting the saturation temperature, for the existing pressure (as shown by a pressure gauge), from the actual temperature of the steam, as shown by a thermometer placed in the steam.

122. Weight of Steam Accounted for by the Indicator Diagram. (a) Not only is it possible to draw a theoretical PV-diagram for a given weight of steam per cycle, as has already been done, but obviously, if a diagram is given and the scale of volumes is known, it is possible to determine the theoretical weight of steam that the given cycle would use. This not only applies to theoretical diagrams, but also to actual ones. The theoretical weight of *dry* steam per *actual* cycle can be found in exactly the same way as for the theoretical cycle. The ratio of what is called the "Steam Accounted for by the Diagram," **"Indicated Steam Consumption,"** or **"Diagram Steam,"** to the steam actually used by the engines is useful in showing the perfection of performance within an engine cylinder. This ratio can be easily obtained, and the difference between the weight of dry steam actually used and the theoretical is the amount liquefied by cylinder condensation.

In the actual case it is convenient to consider the working substance within the cylinder as a mixture of dry steam and water. The indicator diagram shows the behavior of the vapor only.

(b) Suppose the clearance line and zero-pressure line, that is, the PV-axes, have been drawn on an actual diagram, Fig. 91. Then let V_z be the volume as scaled to some point z on the expansion line, between cut-off and release, and let $\mathbf{V}_z$ be the specific volume at the corresponding pressure. Then the total **indicated weight** of *dry* vapor in the cylinder at that time is

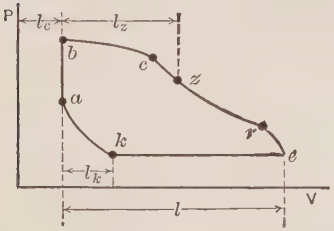


Fig. 91.

$$w_z = \frac{V_z}{\mathbf{V}_z}. \quad . \quad . \quad . \quad (260)$$

Similarly, at any point k on the compression line the weight of *dry* " cushion " steam is

$$w_k = \frac{V_k}{\mathbf{V}_k}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (261)$$

in which V_k is the actual volume as scaled to point k and $\mathbf{V}_k$ is the specific volume for that pressure.

Subtracting the cushion steam from the total in the cylinder gives the **indicated cylinder feed** (w_f') per cycle; thus

$$w_f' = (w_z - w_k) = \frac{V_z}{\mathbf{V}_z} - \frac{V_k}{\mathbf{V}_k}. \quad . \quad . \quad . \quad . \quad . \quad (262)$$

As the quality changes during expansion and compression, the value of w_f' will depend on the locations of points z and k. It is customary to take z either near the beginning or near the end of expansion, and k is usually assumed near the beginning of compression.

(c) Now let y_c = clearance volume ÷ piston displacement per stroke $= \frac{l_c}{l}$ in Fig. 91.

y_z = fraction of stroke completed corresponding to any point on the diagram $\left(\frac{l_z}{l}\right)$.

a = area of piston in square inches.
p_m = m.e.p.
L = stroke in feet.
n = number of cycles per minute.

Then the piston displacement in cubic feet per stroke is $\left(\frac{aL}{144}\right)$

and the volume of vapor when any fraction y_z of the stroke is completed is

$$V_z = (y_z + y_c) \times \left(\frac{aL}{144}\right).$$

Substituting this and a similar value for the volume V_k in Eq. (262) gives for the number of pounds of Indicated Cylinder Feed, per cycle,

$$w_f' = \frac{aL}{144}\left(\frac{y_z + y_c}{\mathbf{V}_z} - \frac{y_k + y_c}{\mathbf{V}_k}\right). \quad . \quad . \quad . \quad (263)$$

Multiplying this by $(60 \times n)$ cycles per hour and dividing by $\left(\frac{p_m Lan}{33{,}000}\right)$, gives the number of pounds of steam per i.h.p.-hr., or **"Diagram Water Rate,"** as

$$W_d = \frac{13{,}750}{p_m}\left(\frac{y_z + y_c}{\mathbf{V}_z} - \frac{y_k + y_c}{\mathbf{V}_k}\right). \quad . \quad . \quad . \quad (264)$$

If points z and k are taken at the same pressure level, p, then $\mathbf{V}_z = \mathbf{V}_k = \mathbf{V}_p$ and

$$W_d = \frac{13{,}750}{p_m \times \mathbf{V}_p}\{(y_z + y_c) - (y_k + y_c)\} = \frac{13{,}750}{p_m \times \mathbf{V}_p}(y_z - y_k). \quad (265)$$

If the pressure is taken as that at the end of compression, then $y_k = 0$ and

$$W_d = \frac{13{,}750}{p_m \times \mathbf{V}_z} \times y_z. \quad . \quad . \quad . \quad . \quad . \quad (266)$$

(**d**) If $W = x_1 \times$ cylinder feed, represents the *dry* steam actually supplied per i.h.p.-hr. and W_d is the indicated water rate corresponding to the point z located at the cut-off, the **"Cylinder Condensation,"** or weight of steam condensed by the cylinder walls and that lost by leakage, is approximately $(W - W_d)$ per i.h.p.-hr.; and the proportion of the whole that is condensed, or **"Condensation Fraction,"** is

$$CF = (W - W_d) \div W. \quad . \quad . \quad . \quad . \quad (267)$$

If the Condensation Fraction, or "Per cent Cylinder Condensation," is known for a certain type of engine under certain operating conditions, then, in considering a prospective engine of this class, the probable water rate under the same conditions can be estimated by dividing the theoretical diagram water rate, found from the probable diagram, by $(1 - CF)$.

CHAPTER XVI.

METHODS OF DECREASING CYLINDER CONDENSATION.

123. Condensation and Leakage. (a) It has already been shown that the cylinder condensation causes the largest loss in the steam engine, with the exception of that inherent in the theoretical cycle. Condensation is evidently dependent on, but not necessarily proportional to, (1) the ratio between the condensing surface (S), to which the steam is exposed, and the volume of steam used per cycle; (2) the temperature difference (T) between the entering steam and the surfaces; and (3) the time (t) of exposure, which is inversely proportional to the number (n) of cycles $\left(\text{i.e., to } \frac{1}{n}\right)$. For computing the probable steam consumption many formulas have been proposed involving functions of S, T, t, or $\frac{1}{n}$, and numerical coefficients determined from experimental data. Such formulas are sufficiently accurate for ordinary purposes, when there is no leakage past piston and valves.*

(b) Unfortunately, while it is possible to determine experimentally whether or not leakage does occur, the *amount* of leakage per cycle cannot be closely evaluated; thus it is impossible to separate the loss due to leakage from that due to condensation. Hence cylinder condensation and leakage must be considered together.

Formulas for cylinder condensation should be derived from a study of data from engines that are known to have little or no leakage. Unfortunately, most of the data available are from engines which were not tested for tightness of valves and pistons and hence are unsuitable for the purpose.

124. Size and Proportions of Cylinder. (a) The **size of cylinder** has an important influence on the cylinder condensation. It can be shown by computation that large cylinders

*For formulas and data see Heck's "Steam Engine," and Thurston's "Manual of the Steam Engine."

have a smaller ratio of surface to volume inclosed than have small cylinders of the same proportions. It is therefore to be expected that large engines will have less cylinder condensation and consequently will give better economy than small ones; and that is actually the case. Very small engines may use twice as much as, or even more, steam per i.h.p.-hr. than very large ones of the same proportions and same conditions of operation.

(b) The amount of **surface in the clearance space** (including that of the steam passages between valves and cylinder) has a predominating influence on the amount of cylinder condensation that takes place; for, just after admission, the piston is moving so slowly that the time of exposure of the steam to these surfaces is comparatively long, hence the amount of condensation that occurs is large. Probably the greater part of the cylinder condensation occurs in the clearance space. The cylinder passages and clearance space should therefore be designed to present the minimum amount of surface consistent with the other considerations involved.

(c) The cylinder condensation is also dependent on the **length of stroke** of the engine. If long and short cylinders are of the same diameter and have their passages and clearance space identically the same, and cut-off steam at the same per cent of stroke, obviously the ratio of clearance surface to the total surface exposed per stroke of the engine is smaller in the long-stroke engine than in the one with shorter stroke. Neglecting the time element, the long-stroke engine should give better economy than the short-stroke one; and in general that is the case, though this is in part due to the fact that the long-stroke engines are usually also larger, have cylinders that are better designed, and have better valve gear than those with shorter stroke.

The time element may have an important influence, however; for example, due to the fact that most of the condensation occurs in the clearance space and because of the shorter time of exposure, some of the short-stroke " high-speed " Corliss engines give as good or even better economy than the long-stroke low-speed Corliss engines.

(d) In many engines the exhaust steam flows over the outer surface of the cylinder wall, on its way to the exhaust pipe.

Because of the high velocity of flow, this steam carries away heat more rapidly than would stagnant air in contact with the same surface. This lowers the mean temperature of the cylinder walls and increases the cylinder condensation. In the better designed engines the exhaust steam is not brought in contact with the cylinder walls after it leaves the exhaust valve.

125. Influence of Point of Cut-off. (a) As most of the cylinder condensation occurs in the clearance space, the later the cut-off (or the greater the volume of steam admitted per cycle), the less will be the *percentage* of steam condensed, although the *amount* may be greater. (1) The percentage of steam not condensed is shown in Fig. 92 (*a*),* by the ordinates, the abscissas being percentage of stroke at cut-off. (2) The work theoretically done per pound of steam *decreases* as the cut-off is advanced in the stroke (because of the reduction of expansion), hence the theoretical steam consumption per unit of work is greater the later the cut-off occurs, as shown by the ordinates of the curve in Fig. 92 (*b*). (3) Evidently, dividing the theoretical water rate per h.p.-hr., Fig. 92 (*b*), by the percentage of steam not condensed, Fig. 92 (*a*), will give the true consumption at the various cut-offs. The values of the actual "water rate," obtained in this way, are shown by the lower curve in Fig. 92 (*c*).

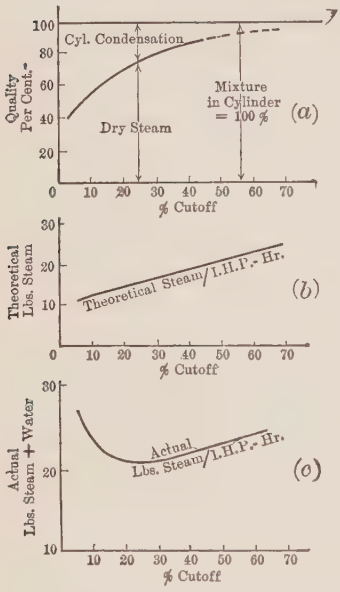


Fig. 92.

Similar "water-rate" curves can be drawn by using data obtained by direct engine test, in which the water per i.h.p.-hr. is measured with engine operating under different loads (i.e., different cut-offs). Usually the water rates are plotted with respect to power output instead of cut-offs.

(b) Inspection of the water-rate curve makes it evident that,

*This is for large four-valve engines having little leakage. See "Engine Tests," by G. H. Barrus.

to give the best economy, *the engine should be operated with cut-off corresponding to the lowest point on this curve.*

The most economical cut-off for noncondensing simple slide-valve engines is about $\frac{1}{4}$ stroke, and for simple Corliss engines it is between $\frac{1}{5}$ and $\frac{1}{4}$ stroke. In practice these are the cut-offs which predominate.

(c) Usually the " water-rate curve " is more nearly horizontal to the right of the lowest point than it is to the left (as in Fig. 92), hence it *is better to " overload " an engine than to " underload " it.*

126. Compounding of Cylinders. (a) By using earlier cut-off the amount of steam used per h.p.-hr. is reduced *theoretically* because of the greater expansion of the steam. But it was seen in the case of the simple engine that cylinder condensation becomes excessive with very early cut-offs because of the greater temperature range and thus defeats the advantage which should be gained theoretically. Therefore, to economically use larger expansions than are possible with the ordinary simple engine, the cylinder condensation must be reduced in some way. It was shown in Section 123 that cylinder condensation can be reduced by decreasing the surface (especially that of the clearance space) to which the high temperature steam is exposed and by reducing the temperature range in the cylinder. Both of these methods can be combined in the following manner.

(b) Suppose a small amount of steam is admitted to a small cylinder (say with $\frac{1}{4}$ the piston area of the simple engine) and that it is expanded only enough to bring the temperature T_R (Fig. 93) of the exhaust steam part way to that of the simple-engine exhaust (say T_R is halfway between T_1 and T_2 on the temperature scale). Let the indicator diagram labelled H.P. in Fig. 93 represent this cycle. Then, owing to the smaller cylinder surface (especially that in the clearance), there is very much less initial and cylinder condensation in this case than if the same weight of steam had been expanded the same amount in the cylinder of the large simple engine.

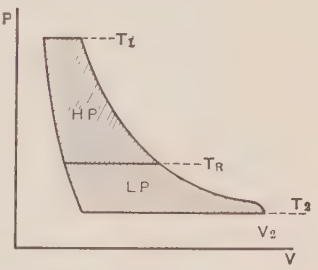


Fig. 93.

Now let the steam exhausted from the small cylinder enter one of the same size as that of the simple engine, and let it be further expanded in this cylinder until the back pressure of the simple engine is reached. The indicated diagram for this case is shown by L.P. in Fig. 93. During this expansion the temperature range (T_R to T_2) is low, hence cylinder condensation is also small here.

(c) It is evident that an engine operating in this manner will use much less steam per h.p.-hr. than will a simple engine; roughly, it uses only about $\frac{2}{3}$ to $\frac{3}{4}$ as much. The best economy with the simple engine is obtained when the steam is expanded in the cylinder to four or five times its initial volume. In an arrangement such as has just been described, the expansion giving the best results is from 7 to 16 or more, depending upon the conditions of operation.

(d) When an engine with two cylinders is arranged to operate in the manner just discussed, it is called a **" Compound Engine "** or " 2X Engine." The small cylinder is named the "*high-pressure (H.P.) cylinder*" and the large one is the "*low-pressure (L.P.) cylinder.*"

Other engines are arranged to expand the steam in three steps, or stages, using in succession three cylinders that progress in size. These are called **Triple-Expansion Engines** (" 3X "), and the cylinders are termed respectively the "*high-pressure*," "*intermediate-pressure (I.P.)*," and the "*low-pressure.*" Triple-expansion engines use considerably less steam per i.h.p.-hr. than do the compound engines.

In the **Quadruple-Expansion Engine** (" 4X "), four cylinders are used in succession. They are termed the *H.P. cylinder*, the "*first intermediate* ($I.P_1.$), the *second intermediate* ($I.P_2.$), and the *L.P. cylinder.* **Quintuple engines** have been made, but their number is very small.

Strictly speaking, the term " Compound Engine " includes all **multiple-expansion steam engines,** but it has become customary to apply it only to those with two cylinders.

Multiple-expansion engines will be discussed more in detail in a later chapter.

A comparison of the performance of simple, compound, and triple engines operating under the same conditions is shown * in

* See report of test, Carpenter, Trans. A. S. M. E., Vol. XVI. Also Thurston, A. S. M. E., XVIII.

Fig. 94. The triple-expansion Corliss engine in the laboratories of Sibley College was tested with high-pressure cylinder operating alone as a simple engine, then with the high and intermediate cylinders acting as a compound engine, and finally with all three cylinders as a triple-expansion engine. The results are shown in

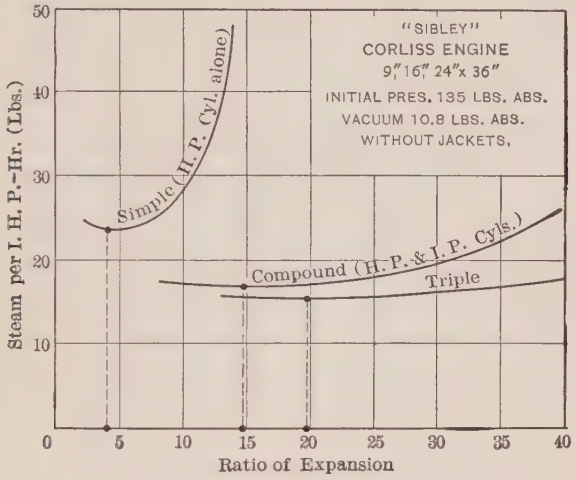


Fig. 94.

this figure. Larger engines and those with jacketing, superheating, etc., would give better results, but this figure shows the relative value of using the different expansions.

(e) **Hirn's Analysis** can be applied to the multiple-expansion engine, each cylinder being considered independently. The data needed for such an analysis (in addition to those required in the case of a simple engine) include the quality and pressure of the steam entering and leaving each cylinder, the weight of condensate "trapped off" from each receiver and the weight and condition of the steam condensed in the reheating coils of the receiver, if such are used. It is then possible to compute the heat in the steam entering and leaving each cylinder and each receiver. Thus besides being able to analyze the heat interchanges and losses of each cylinder considered separately, the same thing can also be done for each receiver, and for the engine as a whole.

127. Gain Due to Condensing the Exhaust Steam. If an engine when operating "noncondensing" (*i.e.*, exhausting to the

atmosphere) gives the indicator diagram shown by the full lines in Fig. 95, with mean effective pressure equal to p_m, then, if the back pressure line is lowered (as shown dotted) an amount equal to p_k pounds, the area of the indicator diagram will be increased as shown, the mean effective pressure will be raised to $p_{mk} = (p_m + p_k)$, and the ratio of the power of the engine to its value when operating non-condensing will be $(p_m + p_k) \div p_m$. Theoretically, however, there will be no change in the amount of steam required nor in the quantity of heat it brings to the engine.

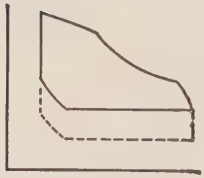


Fig. 95.

By condensing the exhaust steam in a " condenser " (which, being supplied constantly with cold water, acts as a " cold body " in maintaining a low temperature), the pressure of the exhaust steam can be reduced, — and its value will be that corresponding to the condenser temperature. The reduction in pressure below atmospheric may be from 10 to 14 pounds, or even more.

Evidently, in developing the same power, a " condensing engine " would be much smaller than one operating noncondensing, other things being equal. However, owing to the additional cost, operating expense, increased cylinder condensation, and attention involved with a condensing outfit, it is seldom used with simple engines. Multiple-expansion engines, however, are more commonly operated condensing than not.

128. Effect of Superheated Steam. (a) The cooling of the cylinder walls during exhaust is largely due to their surrender of the heat used in evaporating the moisture on their surfaces. As the latent heat of vaporization corresponding to the exhaust pressure is very large, roughly 1000 B.t.u. per pound, the evaporation of a small amount of water results in a very considerable reduction of the mean temperature of the cylinder walls and consequently in an increase in the cylinder condensation when the steam is admitted.

When superheated steam is used there is less moisture in the exhaust steam, and partly because of this, partly because of the slow rate of heat transfer between superheated steam and metal, and partly because the incoming superheated steam can give

up heat without condensing, the cylinder condensation is reduced, and the economy of the engine is improved. Thus by sacrificing superheat to heat the cylinder walls, less heat is required at the boiler for evaporating the water and for superheating the steam used. It is even possible to superheat sufficiently high to prevent all initial condensation. In general, however, it seems probable that superheat should not be so high that the exhaust steam is superheated; although there is some doubt as to this, for one engine test showed better results with exhaust slightly superheated than when just dry.*

In experiments by Ripper† on a small steam engine, it was found that $7\frac{1}{2}°$ F. of superheat would prevent one per cent of cylinder condensation. The specific heat of superheated steam under the test conditions is about 0.53, hence the B.t.u. used in preventing one per cent of condensation was $(7\frac{1}{2} \times 0.53) = 4.0$ per pound of steam. For larger engines and other conditions from 15° to 25° F. and from 8 B.t.u. to 12 B.t.u. per pound are used per percent of saving of condensation.‡

(b) The saving effected by superheating can best be shown by an example:

Let the pressure of steam used by an engine be 135 pounds absolute, for which the latent heat is 870 B.t.u. Then, if it is assumed that 8 B.t.u. superheat will effect a reduction of one per cent in the cylinder condensation, it will save $(870 \times 0.01) = 8.7$ B.t.u. that would otherwise be wasted by cylinder condensation; thus the saving is 1.09 times the expenditure.

If the cylinder condensation is 30 per cent, it could be entirely eliminated if $(8 \times 30) = 240$ B.t.u. superheat were added per pound of steam. Using 0.52 as the specific heat of superheat, the temperature increase would be $240 \div 0.52 = 461°$ F. to effect this result. (Note that this is a much higher degree of superheat than is commonly employed.)

Assume that the boiler furnishes 1000 B.t.u. of heat for each pound of steam generated and that 30 pounds of steam (or 30,000 B.t.u.) are furnished per i.h.p.-hr. Then, since the assumed condensation is 30 per cent, the diagram water rate is

* Carpenter, Trans. A. S. M. E., Vol. XXVIII.

†Superheated Steam Engine Trials, Proc. Inst. C. E. (London), Vol. CXXVIII.

‡ For data and references see Kent's "Pocket Book" and Gebhardt's "Steam Power Plant Engineering," both published by Wiley & Sons.

$(30 \times 0.70) = 21$ pounds of steam per i.h.p.-hr. If the steam is sufficiently superheated to eliminate all cylinder condensation, it will furnish $(1000 + 240)$ B.t.u. per pound; or $21 \times 1240 = 26{,}040$ B.t.u. will be furnished per i.h.p.-hr. Then the economy of the engine is improved in the ratio $\left(\frac{30{,}000}{26{,}040}\right) = 1.15$, while the water rates are in the ratio $(\frac{30}{21}) = 1.43$. Thus it is seen that *the reduction of water rate is not an accurate measure of the saving effected in the heat used.* This example is intended only to show in a very general way the effect of superheat. The numerical quantities for other cases may be very different from those used here.

(c) The saving to be expected by superheating is dependent upon the amount of cylinder condensation that would occur in the same engine if no superheat is used. Evidently the greater this condensation, the larger is the saving possible. Ordinarily the steam consumption is reduced about 6 per cent with 50° F. and about 9 per cent with 100° F. superheat. A reduction of 15 per cent is frequent and as much as 40 per cent has been attained.

(d) It is found that with high temperatures of superheat there is difficulty from warping of cylinder and valves and from failure of lubricants unless they are of the highest grade. A total temperature of 500° F. is about as high as can be used to advantage in ordinary steam engines. Cylinders and valves for higher temperatures should be specially designed for the service. Above 750° F. there is difficulty in finding materials that will endure the temperature for long periods of time.

129. Use of Steam Jackets. (a) Some cylinders are so designed as to be surrounded by "live" steam (usually at high and constant temperature). Such cylinders are said to be "steam-jacketed." Their walls are maintained at higher mean temperature and have less temperature fluctuation than in the ordinary cylinder, consequently there is less cylinder condensation. The heat received by the cylinder wall from the "jacket steam" is the latent heat freed by the condensation of a portion of this steam. If the jacket steam is at the same temperature as the steam entering the cylinder, the mean temperature of the walls will be but little below that of the entering steam, hence the condensation will be small.

At first it may appear that the weight of cylinder condensation thus avoided cannot be more than the steam simultaneously condensed in the jacket, in cases where the condition of the steam entering both the cylinder and the jacket is the same. Because of this, and because the jacket has radiating surface which is larger, and which is maintained at a higher mean temperature, than in the case of the ordinary cylinder, it would seem that no advantage is possible from the use of a steam jacket.

(b) That the steam jacket is beneficial is largely due to the fact that, with its use, the amount of moisture evaporated from the inner walls of the cylinder during exhaust is greatly reduced, thus less heat is abstracted from these walls by the exhaust steam and less steam is used in the cylinder. It has been seen that one pound of moisture evaporated from the cylinder walls carries away roughly 1000 B.t.u. from which there is no return. In the case of the jacket, however, the condensate formed in the jacket can be returned directly to the boiler, and, as it is at boiler pressure and temperature, it will carry back from 250 to 300 B.t.u. per pound. Thus the net result with the steam jacket may be a gain in economy.

In considering the performance of a jacketed engine the heat supplied to the jacket steam must be considered and the water rate must be modified accordingly. If the weight of steam condensed in the jacket per h.p.-hr. is W_j, the heat used per h.p.-hr. by the jacket is $W_j r_j$; and if $(\Delta Q_1 - q_2)$ is the heat added per pound of steam supplied to the cylinder, then the true water rate, supposing the jacket condensate is returned to the boiler without loss of heat, is

$$W = W_c + \left\{ \frac{W_j r_j}{(\Delta Q_1 - q_2)} \right\}, \quad . \quad . \quad . \quad . \quad (268)$$

in which W_c is the weight of steam delivered to the cylinder per h.p.-hr.

(c) As most of the cylinder condensation occurs in the clearance space, this is the most important part of the cylinder to jacket. It is usually only on large engines, however, that the cylinder heads are jacketed, in addition to the cylindrical walls. It would be desirable to jacket the piston, that is, fill it with steam, but practical difficulties prevent this. As there is prob-

ably no advantage from having the exhaust steam superheated, the temperature of the jacket steam should usually not be much higher than that of the steam entering the cylinder. This applies especially in the case of the low-pressure cylinders in multiple-expansion engines.

(d) Steam jackets are not always sources of heat economy. There may be a net loss (1) if they are used with superheated steam, (2) if the cylinder condensation is so small that the jacketing results in superheating the exhaust steam, and (3) if their condensate is not returned to the boiler with little loss of heat. They apparently give smaller returns on large engines than on small ones.

The gain in economy is from 30 per cent down to a negative quantity. Many engineers are skeptical as to their advantage, as the data from various engine tests are somewhat contradictory, and as somewhat greater expense is involved in supplying the jacket equipment.

130. Reheating Receivers. (a) In multiple-expansion engines, it is sometimes the practice to place coils of pipe, containing "live" steam, in the "receivers" through which the steam passes on its way from one cylinder to the next. As the steam in the coils is at relatively high temperature, it superheats (or reheats) the receiver steam, provided the moisture has been properly separated from this latter.

The presence of moisture in the working substance defeats the purpose of the reheating receiver. This moisture should be removed before the steam reaches the reheating coils, for it can be evaporated to better advantage in the boiler.

(b) The action of the reheating coils is similar to that of the steam jacket; and the heat surrendered by the condensation of steam in the coils of pipe is to be charged against the engine.

131. Other Methods of Reducing Cylinder Condensation. (a) Cylinders are always "lagged" with some **nonconducting material** such as asbestos, mineral wool, magnesia, etc., to reduce the radiation. Some small compound "Lokomobile"* engines, which have phenomenal economy, are so arranged that the cylinders are surrounded by the furnace gases as they pass to the stack.

*Herr E. Josse in Zeitschrift des Vereins deutscher Ingenieure, Sept. 12, 1908. Also see report of test of a Wolf engine, (London) Engineering, Oct. 8, 1909.

(**b**) It is of course evident that the higher the **rotative speed,** (or the greater the frequency of cycles), the less will be the cylinder condensation, because the entering steam is exposed a shorter time to the cylinder walls. For example the high-speed Corliss engines use less steam than the low-speed engines of that type, under the same conditions. There are practical considerations, however, which place limits on the speeds of rotation that can be used.

(**c**) It has been seen that *theoretically* the larger the temperature range in the cylinder, the greater is the cycle efficiency. In the actual engine these greater temperature ranges may be obtained by using **higher pressures,** and it has been shown by experiments* that, within limits, there is an increase in economy accompanying their use, even though the cylinder condensation is also increased somewhat.

The gauge pressures (lbs. sq. in.) usual in practice are about as follows:

USUAL GAUGE PRESSURES. TABLE IV.

Simple engines	80 to 125
Compound high-speed engines	100 to 170
Compound low-speed engines	125 to 200
Triple- and quadruple-expansion engines .	125 to 225

(**d**) It has already been shown that the heat economy of the steam engine can be improved by approximating the Regenerative cycle (Section 109). It can also be bettered by selecting the proper compression and the best pressure drop at release, and by reduction of wire-drawing through the throttle valve, the admission and exhaust valves, and the cylinder passages. In some cases, however, the throttling of the steam has been beneficial, probably because the steam at the reduced pressure is superheated a little by the process.

(**e**) By the use of a **Binary Engine** it is possible to use some of the heat in the exhaust steam to vaporize a second and more volatile fluid (such as sulphur dioxide) and to use the resulting vapor in another cylinder from which it is exhausted to a condenser. In this way a considerable increase in power can be

*Gebhardt's "Steam Power Plant Engineering," p. 286, published by John Wiley & Sons.

obtained with the same amount of heat furnished, but at extra expense for equipment, attention, etc.*

(f) If the arrangement of engine is such that as the piston moves it uncovers new portions of the cylinder wall which are at temperatures equal to that which the steam has reached by its expansion, the condensation will be less than in the usual case, in which the steam is brought in contact with walls the whole of which have been exposed, during a considerable period of time, to the temperature of the exhaust steam. Fig. 96 shows

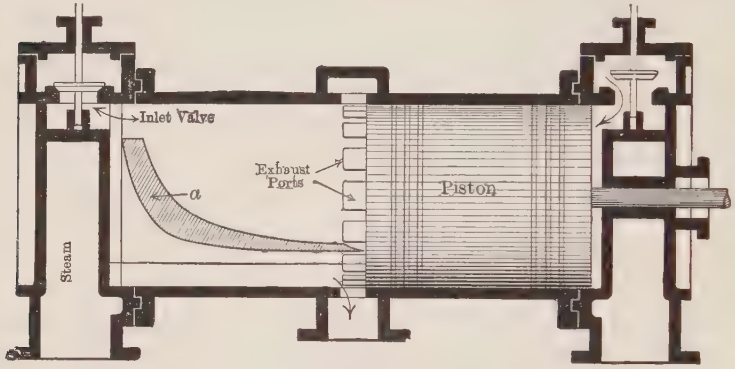


Fig. 96. — Unidirectional-Flow Engine.

a diagram of the **"Unaflow," Straight-Flow** or **Unidirectional-Flow** engine which was recently introduced and which operates on the principle just mentioned. Steam is admitted by the Inlet Valve at the end and is discharged at the middle of the cylinder, when the piston uncovers the Exhaust Ports. As the piston moves from the beginning of its stroke the newly exposed portions of the cylinder wall tend to assume the temperature of the steam with which it is brought into contact; thus there is a gradation of wall temperature from the inlet valve to the exhaust ports. During compression, which comprises practically the whole of the return stroke, the temperature of the steam is raised as the process progresses, and as the volume becomes less the steam is in contact with decreasing surface with increasing mean temperature.

As the expansion proceeds, the steam in contact with the steam-jacketed cylinder head becomes superheated and that in

* See Peabody's "Thermodynamics," p. 280, published by John Wiley & Sons

contact with the piston face is the coldest and contains the most moisture. When release occurs the wettest steam is exhausted and there is little chance for reëvaporation of moisture on the cylinder walls. Exhaust closure entraps the hottest vapor, which, when compressed, attains very high temperature and improves the quality of, or superheats, the entering steam.

Engines operating in this manner have given remarkably good economies, even equalling those obtained with multiple-expansion engines. As great a ratio of expansion is used in the single-cylinder as is employed in the multiple-expansion engine.

(g) When the heat in all the steam exhausted by the engine can be used in drying kilns, in heating systems for houses and factories in winter, or for other purposes, the engine economy is not important, for the heat not utilized by the engine is not wasted. Radiation, conduction, and mechanical friction are always losses, except in cold weather, when they may furnish the proper amount of heat to maintain the right temperature in the engine room; therefore at such times they are not wastes.

CHAPTER XVII.

STEAM ENGINES.

132. Steam-Engine Parts. (a) Fig. 97 shows diagrammatically one of the simplest forms of double-acting steam engine. The various parts of the engine are generally grouped as follows:

(1) **Stationary parts,** — which include the cylinder, cylinder heads (bonnets), steam-chest cover, stuffing boxes, engine frame, outer bearing, and subbase, if used.

(2) **Rotating parts,** — consisting of the shaft, crank (disk), flywheel, and eccentric.

(3) **Reciprocating parts,** — the piston, piston rod, crosshead, and connecting rod.

(4) **Valve gear,** — valve, valve stem (rod), valve-rod guide (or rocker arm), eccentric rod, eccentric strap, and eccentric sheave (or "eccentric").

(b) The steadiness of the rotative speed of the engine during the revolution, that is, during the completion of one cycle on each side of the piston, is controlled by the flywheel. Flywheels will be considered in a later chapter. The number of revolutions, or number of cycles, per minute — which is usually called the engine "speed" — is controlled by the self-acting governor, which in Fig. 97 is of the "throttling," fly-ball type.

The starting and stopping of the engine is controlled by the hand-operated throttle valve which, in special cases, may also be used to regulate the operation of the engine.

(c) Engines usually have the following fittings: drain cocks for cylinder and steam chest; cocks for attaching indicators; lubricating devices for bearings, guides, and cylinders; and shields to collect oil thrown by the crank, the connecting rod, and the eccentric.

(d) Engines are mounted on masonry or concrete foundations sufficiently massive to prevent noticeable vibration being induced in the surroundings. They are fastened to the foundation by "anchor," or "foundation," bolts.

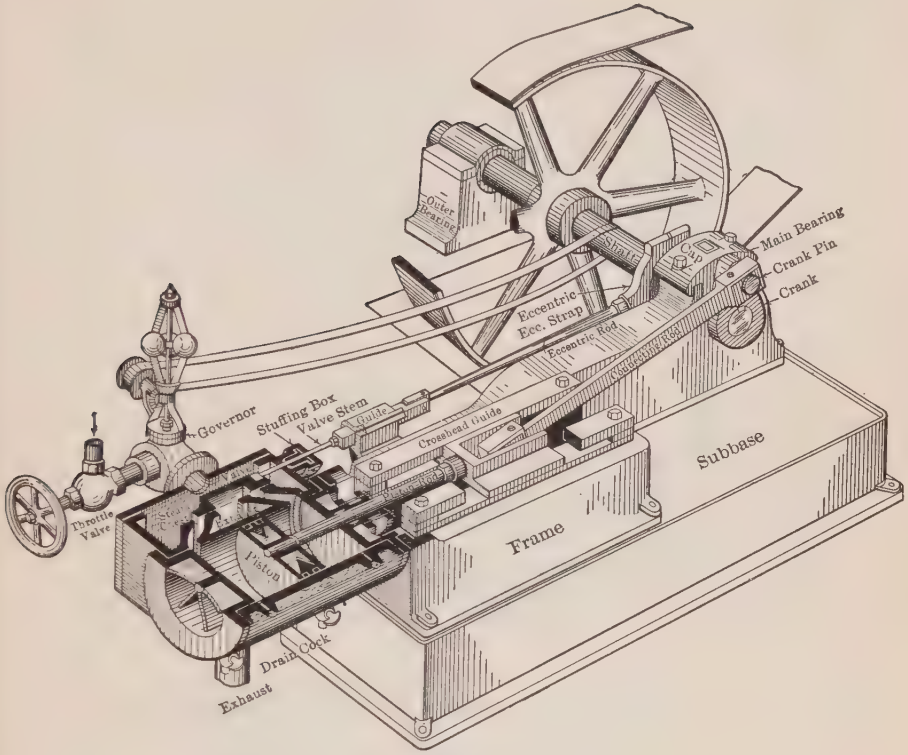


Fig. 97.

133. Classification and Types of Steam Engines. (a) Owing to the great variety of designs and to the diversity of uses to which steam engines are put, it is impossible to give any one classification that would be satisfactory in all cases. The usual **commercial types of stationary engines** are often classified in three groups,—"high-speed," "medium-speed," and "low-speed" engines. By "speed" is meant the rotative speed, when used in this connection.

(b) **High-Speed Engines** are those which have high rotative speeds accompanied by strokes which are very short when compared to the diameter of the cylinder, the piston speed being generally in the neighborhood of 600 feet per minute.* The

* The "piston speed" is the number of feet the piston travels per minute. Thus, if L is the stroke in feet and n is the r.p.m., the piston speed is $V = 2Ln$, since the piston makes two strokes per r.p.m.

stroke is usually about equal to the diameter of the cylinder. These engines almost always have a single "balanced" valve and a shaft governor. They are often called "short-stroke engines," and are designed to occupy the smallest space, have the least weight, and "direct connect" to the smallest dynamo, for a given power, of any of the stationary commercial types. This class includes only engines of comparatively small power, the cylinders usually not being made larger than 20 inches in diameter. Fig. 98 shows a center-crank engine of this type.

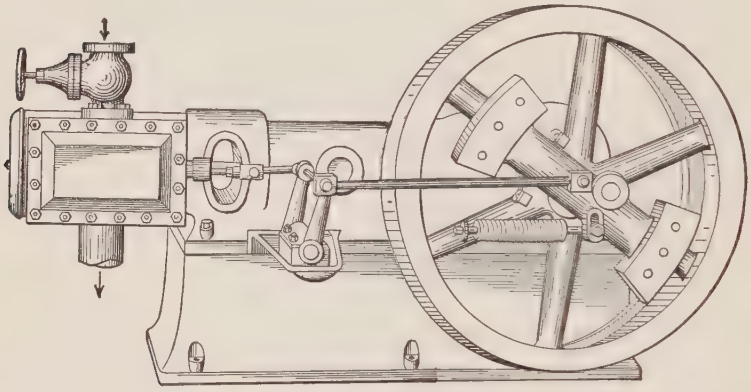


Fig. 98. — Center-Crank Engine with Inertia Type of Governor. (The engine is mounted on a cast-iron subbase.)

(c) **Low-Speed Engines** have long strokes (from 2 to 4 times the diameter of the cylinder) and usually operate at less than 120 r.p.m., the speed being limited by the valve gear, the action of which becomes unreliable at higher speeds. This class includes engines having the "Corliss" and other types of "trip cut-off gear." The governor is usually of the "fly-ball" type. An engine of this kind is illustrated in Fig. 99.

(d) **Medium-Speed Engines** have rotative speeds and strokes intermediate between the foregoing. Positively driven multiple valves are generally used. The cut-off is positive and is often effected by a separate valve. The governor is nearly always of the "shaft type." The piston speed is around 600 feet per minute, being higher on the larger engines. The engine shown in Fig. 100 is of this type.

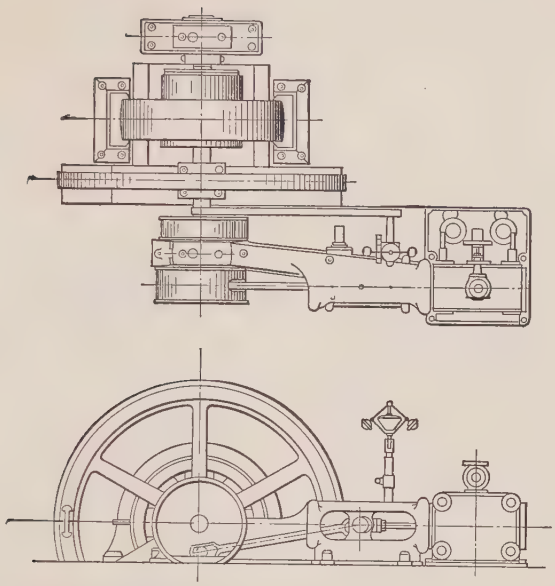


Fig. 99. — Low-Speed Engine with Corliss Valve Gear. Direct connected to an electric generator.

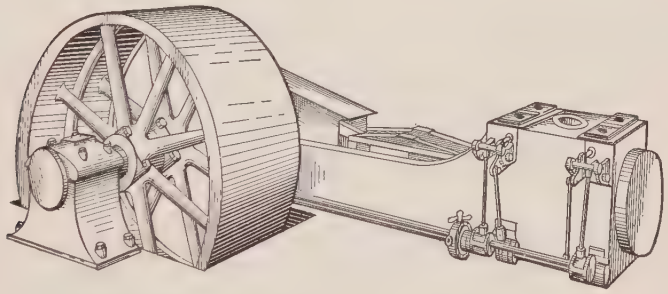


Fig. 100. — Medium-Speed Engine — Shaft Governor — Positive Cut-off.

The medium- and low-speed engines are usually of larger power than the high-speed.

There is no sharp dividing line between these different types of engines, and it is sometimes difficult to decide in which class an engine belongs.

(e) **Vertical Engines** (Fig. 101) occupy less floor space, have smaller foundations, have less cylinder wear, and have slightly greater mechanical efficiency, than do horizontal engines. When

large, they are more difficult to erect, and caring for them involves more effort, as certain parts are reached only by ladders.

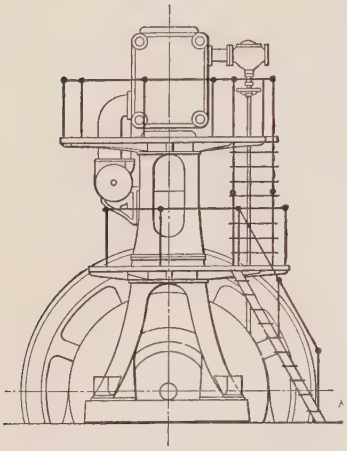


Fig. 101. — Vertical Corliss Engine.

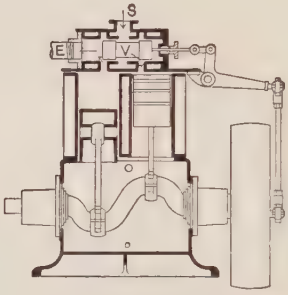


Fig. 102. — Vertical Twin-Cylinder, Single-Acting Engine.

In some special instances engines have been constructed with axis inclined with the horizontal.

(f) **Single-Acting Engines** (Fig. 102) give half as much power as do double-acting engines with the same diameter and stroke of piston, consequently a larger engine is required for a given output. They use pistons of the bucket, or trunk, type, and have no piston rod, therefore they are shorter than double-acting engines.

(g) **Reciprocating Engines** are so called because they have pistons that reciprocate within the cylinder. They are the type that is most common, although engines with rotary piston would apparently be more desirable. Many unsuccessful attempts have been made to devise an engine of the latter form. The difficulty lies in the production of a machine that is economical in the use of steam after the parts have become worn. Prior to 1902 there were issued over 2000 patents on **Rotary Engines,** and none have yet been able to compete with the reciprocating engine as regards steam economy.

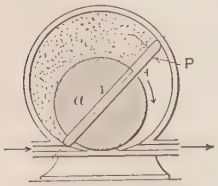


Fig. 103. — Rotary Engine.

In some instances, where small size is more important than operating cost, they may be used to advantage. Fig. 103 shows one of the simplest engines of this type.

In some instances **Oscillating Engines,** such as that shown in Fig. 104, have been used. The connecting rod and crosshead are dispensed with and the shaft is thereby brought closer to the cylinder. Steam is admitted through one trunnion and exhausted through the other. This causes a side thrust, for which proper provision must be made. When the cylinder reaches its extreme position, its inertia causes great pressure to exist between the stuffing box and the piston rod, and these parts must be designed to properly resist this force.

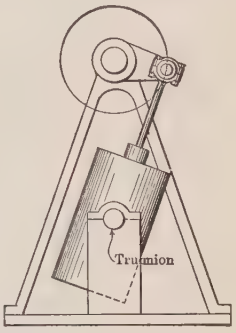


Fig. 104. — Oscillating Engine.

(h) Fig. 105 shows two horizontal **"side-crank engines."** Engines of this type have the disadvantage of having a separate outer bearing which must be aligned with the main bearing. They possess the advantage that there are only two bearings to

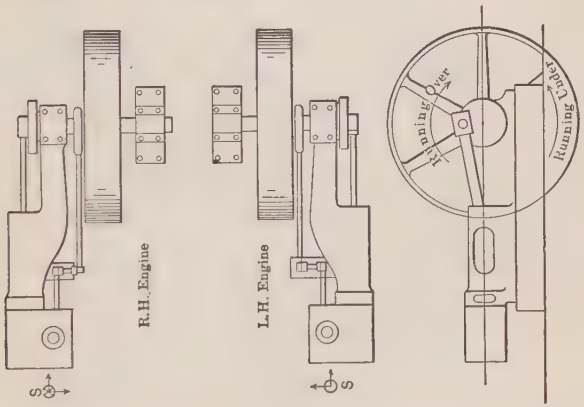


Fig. 105. — Side-Crank Engine — Right Hand and Left Hand — Running Over and Running Under.

be kept in alignment even when the engine is direct-connected to an electric generator, as in Fig. 99.

A horizontal side-crank engine is said to be **right-hand** in arrangement when an observer, standing at the end of the

cylinder (at *S* in Fig. 105) and facing the crank, finds the valve gear and governor parts are to his right; otherwise the engine is "left-hand."

A horizontal engine is said to be **running over** if the crank pin is receding from the cylinder when the crank is above the horizontal center line of the engine; otherwise it "runs under" (see Fig. 105). If a double-acting engine "runs over" the crosshead will exert downward pressure on the guides during both strokes; hence engines are usually operated in this manner.

(i) Fig. 106 shows a **Center-Crank Engine.** In this type the crank is located between the two main bearings, *BB*, and the belt- and flywheels are overhung. If small, these engines may be shipped assembled, ready to be mounted on their foundations. If direct-connected to an electric generator, this latter is substituted for the belt wheel and an outboard bearing is added. There are then three bearings to be kept in line, which is a disadvantage, as very accurate adjustment of bearings is required.

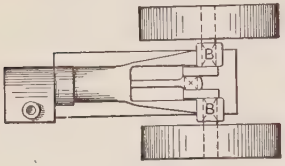


Fig. 106. — Center-Crank Engine.

(j) In some engines the crank case and crosshead-guide chamber are **inclosed** so as to be dustproof and prevent the throwing and waste of oil (see Figs. 102 and 109).

Some inclosed engines are arranged to be **self-oiling** as regards the crosshead, connecting rod, and main bearing. In these cases oil is maintained at such a level in the bottom of the crank case that the crank disk dips into it, and while rotating throws the oil into the crosshead and into collecting pockets from which it is fed to the bearings. The oil then automatically drains back to the crank case and is used repeatedly without being purified.

Other inclosed engines are provided with forced or gravity **oil-feeding systems,** in which the lubricant is filtered each time before reusing.

(k) Compound engines have their cylinders arranged in many different ways.

If the two pistons are on the same piston rod, as in Fig. 107, the engine is called a **Tandem Compound.** Such an engine occupies no greater width than a simple engine of the same power and type, but has greater length. A vertical engine of this type

("steeple compound") occupies the same floor space as the equivalent simple engine.

Either the high- or low-pressure cylinder may be placed next to the frame.

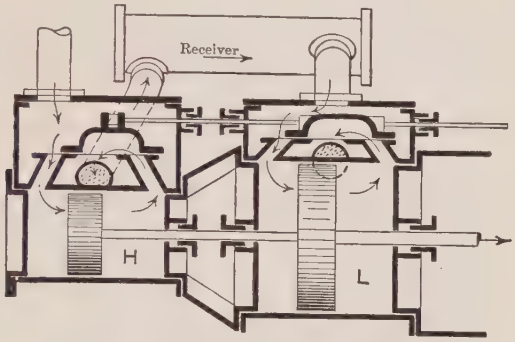


Fig. 107. — Tandem-Compound Cylinders.

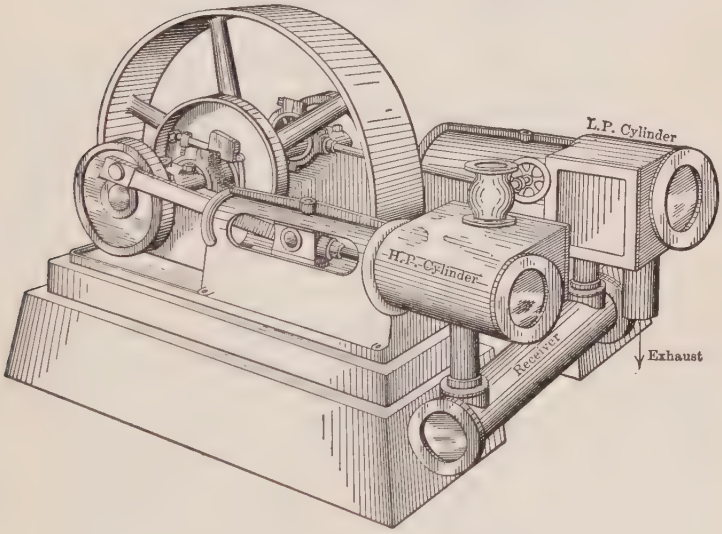


Fig. 108. — Cross-Compound Engine.

If the cylinders are side by side, as in Fig. 108, the engine is called a **Cross Compound.** This engine occupies greater width than the tandem compound, but its length is about the same as that of the simple engine. As it has two frames, and as other

parts are duplicated, it is more expensive than the tandem; but because the cranks may be set at right angles it is possible to obtain greater uniformity of turning effort than with simple or tandem engines, and therefore a smaller flywheel can be used.

In some cases, with this arrangement of cylinders the cranks are placed diametrically opposite (180 degrees apart), but the turning effort is then about as variable as with the single-crank engine.

When the cylinders are immediately adjacent to each other, as in Fig. 109, and have their piston rods attached to the same cross-head, with single connecting rod and crank, the engine is usually

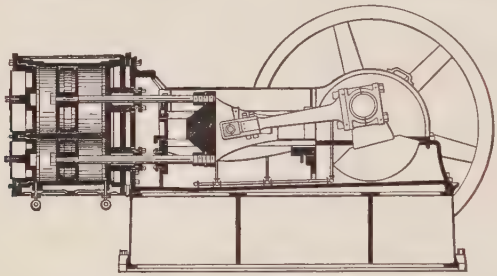


Fig. 109. — Duplex-Compound Engine.

called a **Duplex Compound.** The engine occupies the same amount of space and has the same crank effort as a simple engine.

The arrangement of engine known as the **Angle Compound,** shown in Fig. 110, occupies the same floor space as a simple engine, has the uniformity of crank effort obtained with cranks at 90 degrees (for in this case connecting rods are at 90 degrees and are attached to the same crank pin), and is easily counterbalanced.

In triple- and quadruple-expansion engines the cylinders are arranged in various ways and, looking at the end of the shaft, there may be various sequences with which the cranks pass a given point. The arrangement of cylinders and the sequence and angle between cranks have a predominating influence on the counterbalancing of such engines, as will be seen later when the subject of counterbalancing is discussed.

(1) Engines are used for a great variety of purposes, and are often referred to by their use; thus there are marine engines (Fig. 111), hoisting engines, pumping engines, rolling-mill engines, air-compressor engines, steam-hammer engines, etc.

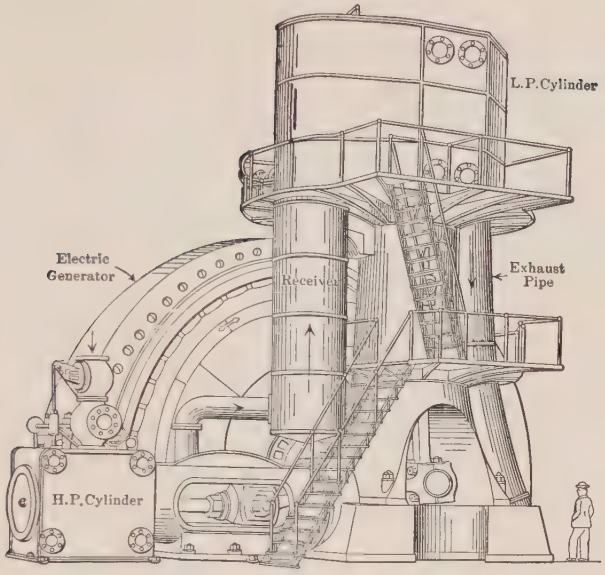


Fig. 110. — Angle-Compound Engine.

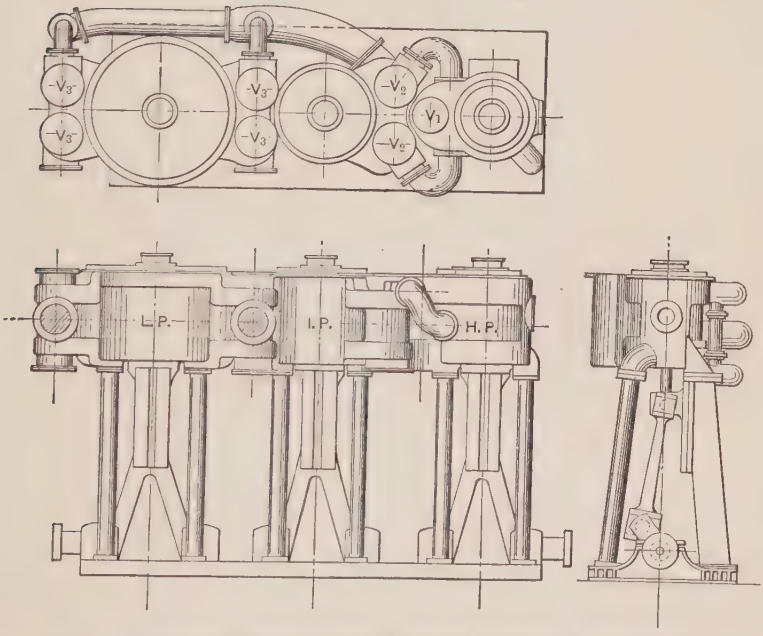


Fig. 111. — Vertical Triple-Expansion Marine Engine — Arrangement.

Engines are also classified as stationary, portable, semi-portable, mobile (marine, locomotive, traction, road roller, and automobile engines).

The uses to which some engines are put require that they be capable of being reversed by hand. This is true of marine engines, some rolling-mill engines, hoisting engines, traction engines, etc. Such engines are called "*reversing engines*" and have special valve gears, either of the "link" or "radial" types, which will be discussed later.

CHAPTER XVIII.

STEAM-ENGINE GOVERNORS.

134. Governing. (a) The term "governing" is applied to the adjusting of the power output or speed of an engine, or both of these, to fit the variable demand.

(b) An engine may be governed in four ways: It may be (1) "hand-governed," as in the case of automobile, marine, and locomotive engines; (2) "mechanically regulated" by a "governor" that acts automatically, as in the usual stationary engine; (3) "resistance governed," its operation being controlled directly by the external resistance; or (4) governed by any combination of these methods.

(c) The ordinary stationary engine is usually **mechanically regulated** to maintain approximately **constant speed** of rotation at all loads. An engine operating uniformly will develop indicated power just sufficient to overcome the friction losses and meet the external demand for power. Should a decrease in the external load occur, it would result in an excess of indicated power, causing an acceleration of the moving parts of the engine, which would continue until the mechanism ruptured under the induced stresses, unless a governor should come into action to prevent. On the other hand, an increase in load would stop the engine unless the indicated power were increased proportionately.

Thus, to maintain constant speed, a "governor" must automatically adjust the indicated power to balance the friction and external load at all times. Exact uniformity of speed is impossible, as a change in speed is necessary to cause a governor to act. This change, however, may be made so small as to be negligible in most cases.

(d) **Resistance governing** is exemplified by an engine directly driving a pump which discharges fluid into a closed reservoir or system of piping. The pump will raise the fluid pressure in the system to the limit of the engine's capacity, when the engine will become ineffective. Should fluid then be withdrawn from

the system, the drop in pressure will cause the engine to start again and to continue running until the pressure once more reaches the limiting value. To prevent the engine from overspeeding in case of sudden withdrawal of fluid, or rupture of pipe, a " mechanical governor " or " safety stop " is usually provided, so adjusted as to automatically come into operation before the safe speed is exceeded. The limit of fluid pressure is also generally made adjustable by hand.

135. Governing of Steam Engines. (a) The adjusting of the power developed within the cylinder to meet the external demand on the engine is usually accomplished in the case of the steam engine either by throttling the steam supplied the cylinder, or by changing the point in the stroke at which cut-off occurs. A combination of both of these methods is possible but is rarely used.

(b) When the engine is **governed by throttling,** the cut-off is fixed by the maximum power which the engine is to develop with steam at the maximum pressure. To obtain less power, the steam is throttled, thus giving lower admission pressure. Fig. 112 shows ideal indicator diagrams for such case.

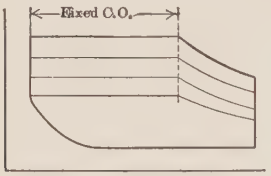


Fig. 112. — Throttle Governing.

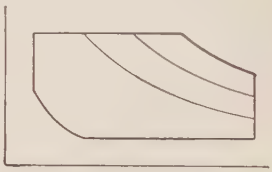


Fig. 113. — Cut-off Governing.

(c) When **governed by changing the cut-off,** the admission pressure is constant, and the amount of cylinder feed is varied, as shown by the ideal diagrams in Fig. 113.

136. Governors. (a) In most cases the demand is for power at *constant rotative speed,* and the governing device should therefore govern " isochronously." Unfortunately, however, governing devices are driven by the engine and operate on a simple mechanical principle which requires a change in speed to make them act. They are also connected to throttling or cut-off devices, each of which must have a different position, or phase relation, for each load. Hence the governors must change posi-

tion with variation of load. As the governor adjustments are brought about only by changes in speed, there is a definite speed and definite governor position corresponding to each load.

Hence the "constant-speed" governor is an anomaly because (1) the governor cannot act until there is a change of speed, and (2) the governor cannot maintain the configuration of valve gear corresponding to different loads unless it runs at different speeds for different loads.

However, with well-designed governors properly adjusted, the amount of variation is small and isochronism is approached sufficiently close for practical purposes.

If n_1, n_2 and n are respectively the lowest, highest and mean r.p.m. of the engine, then the degree of regulation or **coefficient of regulation** is $e = \frac{n_2 - n_1}{n}$, which would of course be zero with isochronous governing.*

(b) The four essentials of a good governor are (1) "closeness" of regulation, i.e., small coefficient of regulation, (2) quickness of regulation, (3) stability or positiveness, and (4) power to move the parts controlled and to resist disturbing forces.

(c) Engine governors may be divided into two classes, — the "Pendulum" or "Fly-ball" Governor, and the "Shaft" Governor.

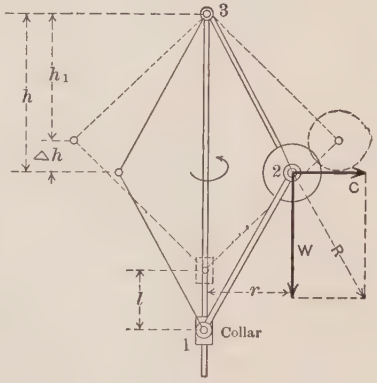


Fig. 114. — Watt Governor.

137. Pendulum Governors. (a) The *simple pendulum, conical, fly-ball* or **Watt governor** is shown in Fig. 114. Corresponding to each different speed at which the vertical spindle and weights revolve, there is a definite height of cone (h) at which the centrifugal force (C) and weight

* It is common practice to speak of the "percentage of speed variation," — thus an engine speed may be said not to vary in excess of $2\frac{1}{2}$ per cent. Such a statement is ambiguous, — by some it is used to refer to the degree of regulation as defined above and by others to refer to the percentage of variation above, or below, the mean speed (i.e., to approximately one-half the degree). Hence the meaning of the term should always be defined if used at all.

of ball (W) will give a resultant (R) which will be in line with the link 3–2, which is the condition for equilibrium. For such conditions the moments of C and of W about 3 must evidently be equal; thus,

$$Wr = Ch. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (269)$$

But

$$C = \frac{W}{g}\omega^2 \frac{r}{12} = \frac{Wrn^2}{35{,}200}, \quad . \quad . \quad . \quad . \quad . \quad (270)$$

in which ω is the angular velocity in radians per second, r is the radius in inches, and n is the r.p.m. Substituting this value of C in Eq. (269) and solving gives for the height of cone

$$h = \frac{35{,}200}{n^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (271)$$

Therefore the height of cone has the same definite value for each rotative speed regardless of the length of arm, the method of suspension, and the weight of the ball. Thus in Fig. 115 the

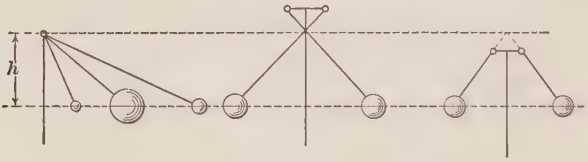


Fig. 115.

heights of cones (to intersection of arms, produced if necessary) are all equal when all weights revolve at the same speed.

(b) The motion necessary to change the steam supply is obtained from the collar of the governor. By referring to Fig. 114, it will be evident that for a given change of speed the collar lift (l) is twice the change in height of cone Δh, when upper and lower arms are equal in length, that is, when the governor is rhomboidal in form.

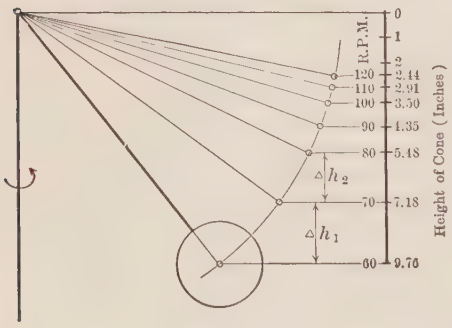


Fig. 116.

Fig. 116 gives the heights of cone for speeds between 60 and 120 r.p.m., advancing by increments of 10 r.p.m. It also shows the different changes in height (Δh_1, Δh_2, etc.), corresponding to

this increment. It is evident that a given amount of collar movement may be obtained with less variation in speed, i.e., with closer "regulation," when the r.p.m. is low than when high; therefore, these governors are usually operated at rather low speeds. This results in their having little "power" to resist disturbances which tend to move the collar, and to overcome resistance offered at that point. Such governors are therefore ordinarily used only when the parts to be moved by the collar are light of weight and practically frictionless.

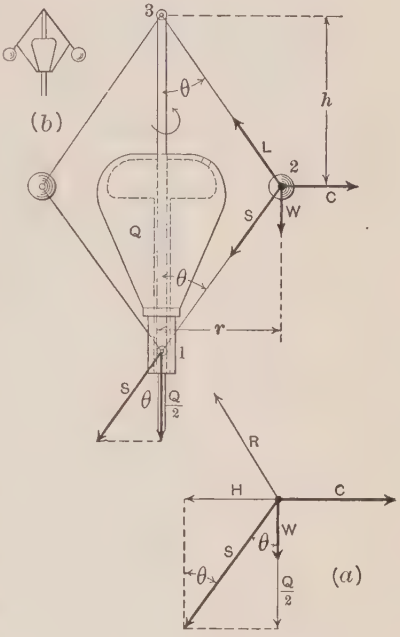


Fig. 117.—Weighted Governor.

(c) The **Weighted Conical or Porter Governor** is shown in Fig. 117 and differs from the Watt governor in the addition of the central weight Q which rests on the collar, and for a given speed causes the weights to revolve in a lower plane (i.e., with height of cone greater) than in the simple form. Evidently, within limits, a given height of cone can be had at any speed by merely placing the right amount of weight at Q.

At the collar, in the figure, is drawn the triangle giving the component S of $\frac{Q}{2}$ along the link 1–2. The ball is subjected to forces S, C, W, and a resultant tension L (in link 3–2) which must point toward the pivot 3. These forces are shown at (a) in the figure, from which it is evident that the vertical component of S is $\frac{Q}{2}$ and the horizontal component H is $\frac{Q}{2}\tan\theta = \frac{Q}{2}\frac{r}{h}$, since $\frac{r}{h} = \tan\theta$. For equilibrium the moments about 3 of the horizontal and vertical forces or components must be equal, and therefore (remembering that L passes through 3),

$$\left(C - \frac{Q}{2}\frac{r}{h}\right)h = \left(W + \frac{Q}{2}\right)r.$$

Substituting the second value of C from Eq. (270) and solving for the height of cone gives*

$$h = \left(1 + \frac{Q}{W}\right)\frac{35{,}200}{n^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (272)$$

Comparison of this equation with Eq. (271) will show that it is possible for the loaded governor to have the same height of cone, and same *degree* of regulation for a given collar movement, with high speeds, that can be obtained with the simple conical governor with low speeds only. The " loaded " governor is much the more powerful of the two because of this fact.

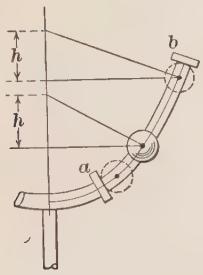


Fig. 118.

(d) Eqs. (271) and (272) show that to have **isochronous governing** (n = constant) the height of cone (h) must be constant. Thus in Fig. 118 the path of the ball must be such that the sub-normal to the curve is constant. As this is the property of the parabola, the ball should be guided over such a path for this kind of governing. In such case, at the given speed, the ball would be in equilibrium at any and all points on the guide; that is, the forces would always be in equilibrium.

Such an arrangement is, however, of no commercial value because (1) if a disturbance increased the speed slightly the equilibrium would be destroyed, the centrifugal force would predominate, and the ball would seek the extreme position against the outer " stop " b; (2) a decrease of speed would cause the weight to move to the inner stop a; and (3) there is no definite place for the ball at the speed of isochronism — it is balanced at any position on the guide; whereas to be of practical value there must be a definite position of ball corresponding to each different load to which the engine is subjected. Hence, while this governor is ideal as regards constancy of speed, it is **unstable** and of no commercial value. It serves as a limit which actual governors may be made to approach as closely as is possible without introducing instability.

(e) Fig. 119 shows an arrangement in which the path of the ball is a circular arc approximating the parabolic path, but de-

* Note that this equation applies only when the governor linkage is rhomboidal in arrangement. If the arrangement is like that at (b) in Fig. 117 the formula must be modified.

parting therefrom somewhat in order to have the difference between h_2 and h_1 sufficient to insure stability. Such an arrangement is described as a governor with *crossed arms*.

With the governor previously described, which has the suspension point located on the spindle, the path of the ball departs

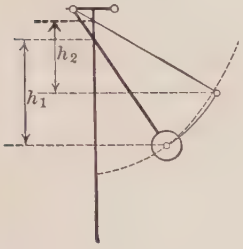


Fig. 119.

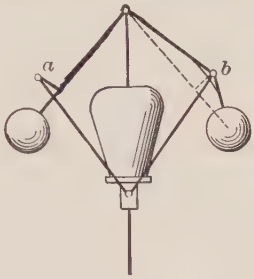


Fig. 120.

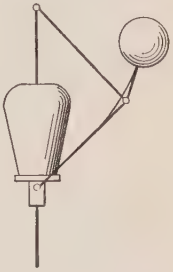


Fig. 121.

widely from the parabola, hence such governors do not give close regulation except when the collar movement is small.

(f) Isochronism can be approached by having the point of attachment between links *offset* from the weight arm, as at *a* in Fig. 120, or using the equivalent *bent arm*, as at *b*. The theory of this type of governor will not be included here.*

Fig. 121 shows another arrangement of governor with which isochronism can be approached.*

(g) Eqs. (271) and (272) apply only in the ideal case in which there is no resistance to be overcome. If the collar and pins have friction, the speed of the governor must change a considerable amount, Δn, before the centrifugal force is changed by a sufficient amount to enable it to overcome this resistance and cause the collar to start to move. Thus, if the change in speed necessary to overcome the resistance when the weights tend to move out equals that when the tendency is inward, there can be a change in speed equal to $2\ \Delta n$ without movement of the collar. Evidently the degree of total regulation is similarly affected. The greater the collar friction the larger the influence, hence the collar friction, the resistance of all parts moved by the collar, and the friction of the governor parts must be as small as possible, if close regulation is desired.

* See Tolle, "Die Regelung der Kraftmaschinen," Julius Springer, publisher, Berlin.

138. Spring-balanced Fly-ball Governor. In the fly-ball governors so far discussed the moments of the centrifugal forces acting on the balls were balanced by the moments of the gravity forces. Some governors are so arranged that the centrifugal forces (or their moments) are practically entirely balanced by one or more springs, as in Fig. 122. In other governors the centrifugal force is balanced by a combination of gravity and springs.

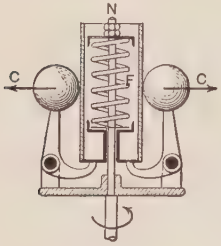


Fig. 122.

The degree of regulation of the governor shown in Fig. 122 can be adjusted by means of the nuts N, which can be used to change the initial compression of the spring. This governor operates on the same principle as the simple centrifugal "shaft" governor described in the next section. There are many different arrangements of governors of this type. They may be made to operate at high speed and have considerable power, and they can be adjusted to give "close" regulation.

139. Elementary Shaft Governors. (a) The shaft governor is so called because it is mounted either in the flywheel or in a governor case carried by the main shaft of the engine.

(b) The elements of the simplest form of this governor are shown in Figs. 123 to 125. Referring to Eq. (270), it is seen that

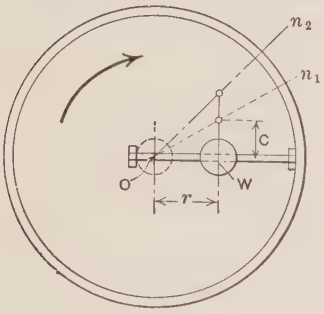


Fig. 123.

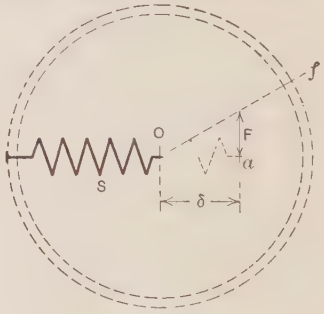


Fig. 124.

if the speed of rotation is constant the centrifugal force C with a given weight varies directly with the radius r. Thus in Fig. 123, with constant speed = n_1, the ordinates (C) of the line On_1 show the manner in which the centrifugal force increases as the weight

W is moved outward from the center of the shaft. On_2 shows the same thing for a higher constant speed n_2, and similar lines could be drawn for each other speed.

(c) In Fig. 124, S is a spring with end at O when free. The ordinates (F) of curve Of show the increase of the spring force with the elongation δ. As the curve is similar in character to those in Fig. 123, it would be possible to place the spring in the flywheel, with end O at the center of rotation, and thus cause Of to coincide with one of the On-curves. The centrifugal force would then be balanced by the spring pull in all positions of the weight, for that particular speed. Hence this arrangement would give isochronism. The speed at which this equilibrium occurs depends of course on the strength of the spring.

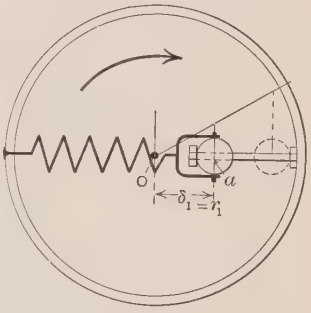


Fig. 125.

(d) If a in Fig. 125 is the position of the center of the ball when against the inner stop, then when the weight is in the "inner position" the spring will have elongation equal to Oa. The extension of the spring with the weight in the inner position is called the "initial elongation." In adjusting the spring to give isochronous governing, the initial elongation δ_1 must be equal to the distance Oa, which is equal to r_1.

(e) The isochronous shaft governor is an unstable one; any change from the speed of isochronism causes the weight to move to one or the other extreme position; and at the speed of isochronism the weight is in equilibrium at any position in its path, that is, has no definite position. Therefore, this governor is of no commercial value, but is the limit which actual governors may be made to approach as closely as is possible without introducing instability of action.

(f) In order to have **stable governing,** there must be definite positions of the weight W for each different load on the engine. If the end of the spring, when not under tension, is at o', to the right of the wheel center o in Fig. 126, instead of at o, then the line $o'f$ will cross the on-curves; point 1 will correspond to a speed equal to n_1, point x' to n_x, and point 2 to n_2; thus, when the weight is at a the spring pull will be balanced by the centrifugal force

when the speed is n_1, at x there will be equilibrium if the speed is n_x, and at b the forces are equal when the speed is n_2.

With such an arrangement there is a definite position of the weight at each different speed, thus the arrangement is stable.

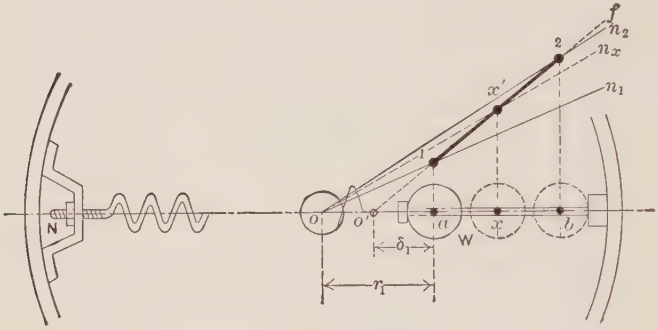


Fig. 126.

If the position of W fixes the power developed by the engine, then there will be a different speed for each different power output.

The line *1–2* is sometimes called the Characteristic Curve (C-curve), as its position with respect to the constant speed curves, *On*, indicates the character of the governing.

(g) If a is to be the inner position of the weight, and n_1 the lowest speed, then point *1* is fixed. The speed corresponding to position b is determined by the slope of the Characteristic Curve *1–2* relative to the constant-speed curves, and is dependent on the distance oo' or on the distance δ_1. If o' coincides with o, the initial elongation $\delta_1 = r_1$, and the governor would be isochronous, as the C-curve would coincide with on_1. The greater the distance oo' is, i.e., the smaller the initial elongation δ_1 (compared with r_1) the greater will be the speed variation between limits a and b, and the greater will be the stability, and *vice versa*.

(h) The adjustment of a governor is divided into two parts as follows: — (1) The initial elongation of the governor spring, would be increased (by means of nut at N in figure) until the best degree of regulation that is consistent with stability is obtained. Note that the degree of regulation is dependent only on the amount of the initial elongation, and that it is independent of the strength of spring and of the weight of ball.

(2) After the spring has been adjusted to give the proper degree of regulation, the speed can be changed to any desired value, within reason, by changing the weight W. If the weight is reduced the speed will increase until the centrifugal force balances the spring pull; if increased, the effect on the speed will be the opposite.

(i) In Fig. 127 the weight W is mounted on an arm pivoted at 3 and with spring S attached at 1. If it is considered that the arc ab here approximates path ab in Fig. 126, the spring S would have initial elongation δ_1, the same as in that case. Evidently spring S can be replaced by spring S' if the latter is made l/l' times as strong (W remaining the same) and if the initial elongation is made equal to $(l'/l)\delta_1$.

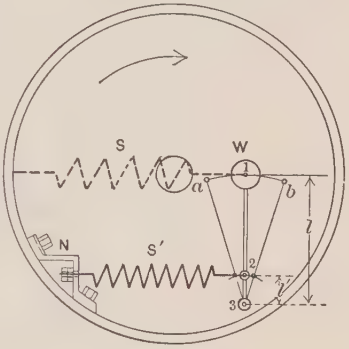


Fig. 127.

This arrangement contains the elements of the more common forms of commercial shaft governors. It is adjusted in the same manner that was described in (h) for the simple case.

140. Commercial Types of Shaft Governors. (a) In general, the commercial shaft governor has one, or two, pivoted "weight arms," the centrifugal force acting on which is balanced by one or more springs which are so adjusted that there is a different speed and a corresponding definite and distinct position of the arm, or arms, for each different load on the engine. The "weight arms" are connected either directly, or by links, to the eccentric, so that for each speed there is a definite and distinct position of the eccentric, a corresponding cut-off, and a definite amount of power developed. If the load changes, the speed of the engine will also change until a cut-off is found which gives the right amount of power to meet the demand.

If the governor is of good design and properly adjusted, the total amount of speed variation is very small (being from 1 to $2\frac{1}{2}$ per cent of the "normal" or average speed); thus the speed is practically constant.

There are two general types of shaft governors, —the "Centrifugal" and the "Inertia."

(b) The Sweet governor, which was one of the earliest of the **centrifugal governors** and which is still widely used, is shown in Fig. 128. Pivoted to one of the arms of the flywheel, or governor wheel, is a "weight arm," which has a heavy head *W*. When the engine is not running, this weight arm is held in the "inner" position (that shown in full lines) by the leaf spring *S*. After steam is turned on, the arm will remain in this position until the speed has reached a certain point (for example, say 198 r.p.m.),

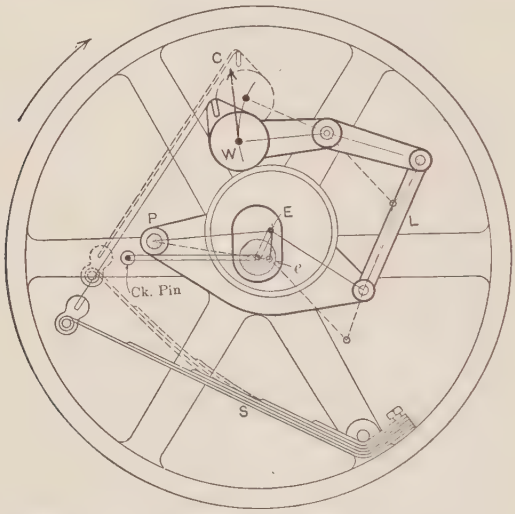


Fig. 128. — Sweet Type of Centrifugal Governor.

when the centrifugal force *C* will just balance the spring pull. If the speed is raised further, the increased centrifugal force will cause the arm to move outward until, at some speed (say 202 r.p.m.), it reaches the extreme "outer" position (that shown by the dotted lines). At the "normal" speed (200 r.p.m.) the weight arm would be about midway between these extreme positions; and for every other speed (between the 198 and 202 r.p.m.) there are definite positions of the arm.

In the example the total variation in speed is 2 per cent of the normal r.p.m. By changing the adjustment of the spring, however, the amount of variation can be altered, but if it is made too

small the friction and inertia of the valve gear, and the other disturbances, will make the action of the governor uncertain, — so there is a practical limit to the closeness of regulation.

Again referring to Fig. 128, it is seen that the arm carrying the eccentric is pivoted at P to one of the arms of the wheel, and is connected by a link L to an extension of the weight arm. When this latter is in the inner position, or is "in," the center of the eccentric is at E, the position for the latest cut-off; and when it is "out," the eccentric center is at e, the position for zero cut-off.

The manner in which the governor operates is as follows: When the engine is standing still, the governor holds the eccentric in the position E for the latest cut-off. When steam is turned on, the engine will speed up until a certain r.p.m. is reached, at which the governor arm will begin to move out, thus shifting the eccentric towards e and decreasing the cut-off. This movement will continue until a position is reached at which the power developed just equals the load, and as long as this latter remains constant the governor arm will remain in this position. Now, if the load is reduced, the engine will speed up (tending to run away), and this causes the weight arm to fly out, shifting the eccentric nearer to e and reducing the power developed until it becomes again equal to the demand. Similarly, if the load is increased, the speed of the engine will decrease, and, as the weight arm moves "in," the cut-off will be increased, until at some position of the arm a balance is again reached between the power and the load.

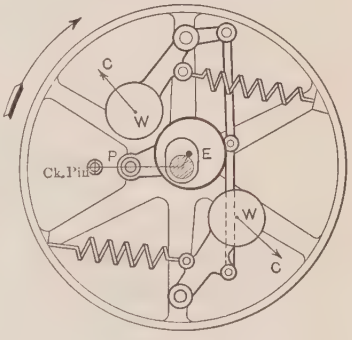


Fig. 129. — Centrifugal Governor.

Fig. 129 shows another "centrifugal" shaft governor; but in this case there are two weight arms, symmetrically placed, instead of one. In its action, this governor is identical with that which has just been described.

(c) Fig. 130 shows the Rites **Inertia Governor**, which consists of a long weight arm (WW'), an eccentric pin E, and a spring. The arm is pivoted at P, close to the shaft, and its end W' is heavier than W, so the center of gravity is at G. The position of

the parts shown in full lines is for latest cut-off, and is the one occupied when the engine is not running; that shown by the broken lines is for the earliest cut-off. In the former position, the arm is said to be " in," and in the latter, " out." The direction of rotation is shown by the arrow. The governor will not operate satisfactorily if the direction of rotation is reversed without making changes in the governor itself.

As the engine starts up, the governor arm remains in the inner position until a certain speed is reached, when the centrifugal

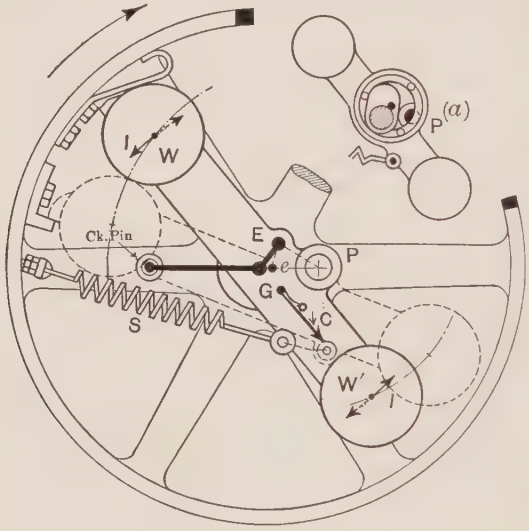


Fig. 130. — Inertia Governor.

force C, acting on the weight arm, becomes sufficiently great to balance the spring pull. Then, with a further increase in speed, the weight arm will move out (the eccentric meanwhile moving toward e) until a sufficiently early cut-off is obtained.

Now, if the load falls off the engine will speed up, and the increased centrifugal force will cause the weight arm to move out until the cut-off is reduced to the proper amount, the action being just the same as in the case of the centrifugal governor. However, in addition to the centrifugal force acting on the arm, there is also an inertia force which assists the movement.

The inertia of the weight arm acts in this manner: As the engine speeds up the governor arm tends to continue to rotate

at its old speed, because of its inertia, and hence lags behind the wheel, moving with respect to the latter in the direction shown by the arrows *I* and *I* in the figure. It is seen that this movement is in the same direction as that caused by the centrifugal force *C*. Again, if the load is suddenly increased the engine will slow down, but, because of its inertia, the weight arm will continue at its old speed, thus gaining on the flywheel, and again assisting the centrifugal force in changing the position of the eccentric and weight arm with respect to the crank.

It is seen that the inertia governor is primarily a centrifugal governor, but that, in addition, the weight arm is so pivoted, and has its weight so distributed, that its inertia assists in making the adjustment, and that the more sudden the change in the load the greater will be the assistance it renders.

In this form of governor the eccentric, or eccentric pin, is usually mounted directly on the weight arm. Sometimes the eccentric is keyed directly to the fulcrum pin on the end opposite that to which the arm is fastened. With these arrangements, in order to have the inertia of the weight arm act in the right direction, the *fulcrum pin must be placed on the side of the shaft opposite to the crank pin*, when an "external" valve is used, and on the same side when the valve is "internal" (Section 143).

On "center-crank" engines the governor is frequently placed in the outer side of the wheel, in which case, since the shaft does not extend beyond the governor wheel, the arrangement can be that shown in Fig. 130. If, however, the governor is placed on the side of the wheel next to the engine frame, both the governor arm and the eccentric must be made to surround the shaft in the manner shown at *a*, Fig. 130.

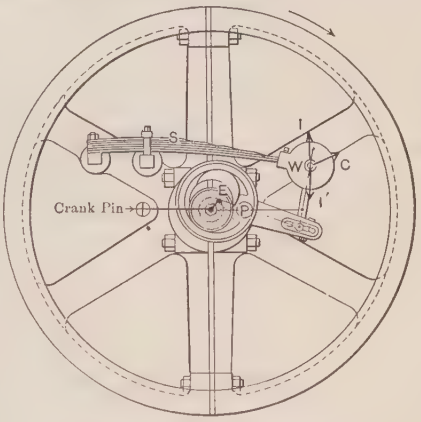


Fig. 131. — Armstrong Governor.

Fig. 131 shows the Armstrong governor, which is of the inertia type. The weight *W* is mounted on the end of the leaf spring *S*, and is subjected to cen-

trifugal force C, and also to inertia force I or I' when sudden change occurs.

(d) For both forms of shaft governors it has been seen:

(1) That there is a definite speed, cut-off, and power for each position of the weight arm.

(2) That when the arm is "in," the speed is the lowest and the cut-off is the latest; whereas, if the weight arm is "out," the reverse is the case.

(3) That an increase in load decreases the speed and causes the arm to move "in," which gives a later cut-off; whereas, the effect of a decrease in load is the reverse.

(4) That, for close regulation, the friction and inertia of the valve-gear parts must be small, and especially is this necessary when the inertia form of governor is used.

(5) The adjustments of spring to obtain the desired degree of regulation, and of weight to obtain the speed wanted, are made in the manner outlined in Section 139 (h) for the elementary governor.

There are almost an unlimited number of forms of shaft governors, but all of them are merely modifications of those which have been described.

CHAPTER XIX.

THE VALVE GEARS OF STEAM ENGINES.

141. Introduction. It is assumed that the reader is already familiar with the arrangement and operation of the simple steam engine having the plain slide valve, and that he is able to use at least one kind of "valve-gear diagram" for the analysis or design of a simple "D-valve." The purpose of this chapter is mainly to review certain definitions, to bring out certain conceptions which will be useful in the later discussions, and to give a brief discussion of the different types of valve gears used on steam engines.

142. The Engine. Definitions. (a) The *crank end* (C. E.), or front end, of the cylinder, or valve, is the one nearest the crank,

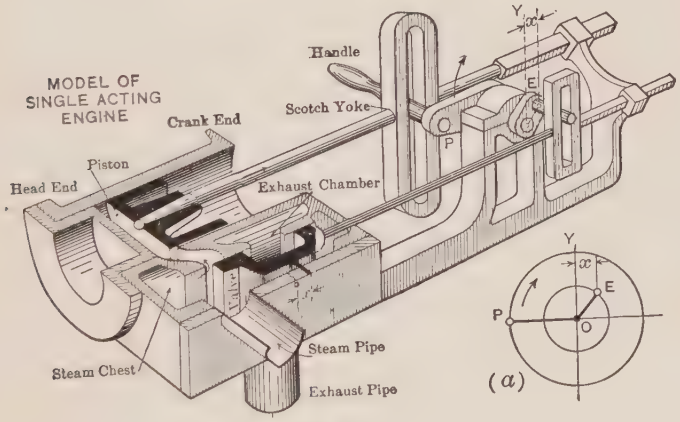


Fig. 132.

or next to the engine frame. The opposite end is the *head end* (H. E.), or back end.

(b) The *forward* (Fd.) *stroke* of the piston or valve is that towards the crank. The return stroke is the *back* (Bk.) *stroke*.

(c) Fig. 132 shows a model of a simple single-acting engine with piston and valve driven by crank and eccentric pins operating in *Scotch yokes*, or *slotted crossheads*. It is evident that with this arrangement the valve and piston will have simple harmonic motions, and in consequence the analysis of the valve action is a simple matter.

The motions, with this arrangement, are exactly the same as would occur if the engine had connecting and eccentric *rods of infinite length*.

(d) Steam engines, of course, have connecting rods and eccentric *rods of finite length*, and the "angularity" of these rods causes the motions of piston and valve to depart slightly from the true harmonic. The eccentric rods are usually so long, however, when compared to the radius of the eccentric crank, that the departure in the case of the valve is negligible. If the analysis of motions is to be only closely approximate, the motion of the piston may also be taken as true harmonic, which simplifies the problem.

(e) The crank is on *dead center* when the piston is at the end of the stroke, and is then horizontal on horizontal engines. When the piston is at the head end of the cylinder, the crank is on the "head-end dead center"; when at the other end, it is on the "crank-end dead center."

(f) The *eccentric* (ecc. or E) is really a crank with pin of such large diameter as to surround the shaft. In the following discussion the term "eccentric" will be used as applying to the center of this pin. Like other cranks, the eccentric has dead-center positions.

(g) The *throw* of the eccentric is the "eccentricity" or length of the crank. (There is a lack of agreement in the use of the term "throw," some using it in the sense given and others as meaning the total movement of the valve or "travel.")

143. The Valve. Definitions. (a) Fig. 133 shows the longitudinal section of a simple D-valve suitable for a single-acting engine which takes steam at only the head end of the cylinder. This valve is arranged to admit steam to the cylinder past the left *outer edge* (when *steam edge* s of the valve passes to the right of the steam edge S of the port), and to exhaust the steam from the cylinder past the left *inner edge* (when the *exhaust*

edge u of the valve moves to the left of the exhaust edge *U* of the port).

(b) The *width of the port* is the distance *SU* in the figure.

(c) The valve shown is called an *external valve*. If the valve admitted steam to the cylinder past its inner edge and exhausted at the end, it would be an *internal valve*, and would have to be of different design from that shown here. Unless otherwise stated, the valve will be assumed to be external.

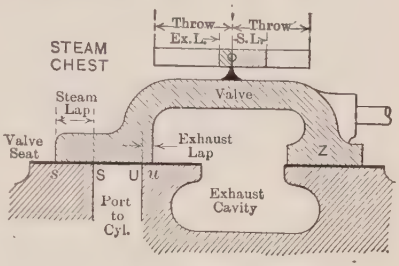


Fig. 133.

(d) The terms "*steam chest*," "*exhaust cavity*," and "*valve seat*" should not need explanation (see Fig. 133).

(e) The *valve is central* (or in *mid-travel*), with index at *O* in Figs. 132 and 133, when the eccentric is vertical, either up or down.

(f) The *lap* of the valve is the distance between the valve edge and the port edge with which it operates, when the valve is central. The *outside lap* (or outer lap) is that of the outer edge, and the *inside lap* is the lap of the inner edge of the valve. The *steam lap* (S. L.), see Fig. 133, and the *exhaust lap* (Ex. L.) are respectively those of the steam and exhaust edges of the valve. The lap is positive if the port is closed when the valve is central and negative if open. (Negative lap is sometimes called "clearance.")

(g) The *valve opening* is variable and is dependent on the displacement of the valve; but the term is usually understood as referring to the maximum *width* of the opening unless otherwise stated.

(h) The *travel* of the valve is the *stroke* or total amplitude of its motion. If the valve is direct-driven, the travel is equal to the diameter of the eccentric circle.

(i) The term "*displacement*," when applied to the valve, will be understood to mean the distance the center of that part has been moved from its central position; and in the case of the eccentric on a horizontal engine it will be the horizontal distance from the center to the vertical center line of the shaft.

(j) The **four periods of operation** of the valve are admission, expansion, exhaust, and compression.

(k) The **four principal valve events** are admission (A), cut-off (C), release (R), and compression (K). The **four minor events** are maximum displacement of the valve to the right (M), same to the left (m), valve central and moving to the left (Q), and central but moving to the right (q).

The letters given in the parentheses in the above list will be used to indicate the respective events on the diagrams which are to follow.

(l) *Unless it is specifically stated to the contrary, it will always be assumed in the following discussion that the engine is horizontal, with cylinder to the left of the crank shaft, that an "external valve" is used, and that the crank rotates in a clockwise direction.*

144. Action of the D-Valve and Eccentric. (a) When the valve is driven by a Scotch yoke, it is seen, by referring to Fig. 132, that (1) the valve is central when the eccentric is on the vertical center line OY through the center of the shaft, (2) the valve will be in this position whether the eccentric OE is vertical upward or downward, and (3) that at all times the displacement x of the eccentric E equals the displacement x of the valve.

(b) In Fig. 133 it is seen that the valve must be displaced to the right a distance equal to the steam lap before opening to steam occurs, and that any further displacement represents opening. In Fig. 134, in which the radius of the circle equals the eccentric throw, the distance from the eccentric center (anywhere on this circle) to the axis qQ is the valve displacement. On this figure the "*steam-lap line*" AC has been drawn at a distance equal to the steam lap to the right of qQ. Hence the steam edge is open an amount equal to the horizontal distance the eccentric is to the right of line AC. In Fig. 133 it is seen that the valve must be displaced to the left a distance equal to the *exhaust lap* before exhaust opening occurs, and that further displacement in that direction represents the amount of opening. In Fig. 134 the *exhaust-lap line* KR is at a distance which equals the

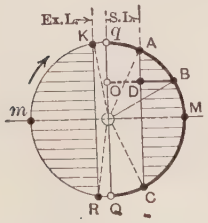


Fig. 134.

exhaust lap to the left of Qq. Hence the exhaust edge of the valve is open an amount equal to the horizontal distance that the eccentric is from KR, if it is to the left of that line. If the exhaust lap is negative, KR will be to the right of qQ.

(c) Starting with the eccentric center at q in Fig. 134, the valve is evidently central and moving to the right, since the rotation is clockwise. When the eccentric reaches A, the valve displacement equals the steam lap, and *admission* occurs; when at B, the valve is displaced a distance BO to the right and its steam edge is open an amount equal to BD; at M the valve has maximum displacement and maximum steam opening; at C the valve, now moving to the left, has displacement equal to the steam lap, and *cut-off* is occurring; at Q the valve is central and its edges are overlapping by amounts equal to the respective laps; at R the valve is displaced to the left an amount equal to the exhaust lap, and is opening to *release;* at m the valve displacement is maximum to the left, and maximum exhaust opening is reached; and at K the displacement is equal to the exhaust lap, so that exhaust closure or *compression* is beginning. The amounts of openings to steam and exhaust are shown by the lengths of the horizontal section lines. Fig. 134 may be called a **rectilinear diagram of valve displacements.**

(d) Note that admission and cut-off are controlled by the same valve edge (steam edge) but with valve motions opposite. This is apparent not only from Fig. 133, but can be seen from line AC in Fig. 134. Similarly, compression and release are controlled by the same edge (exhaust edge). Valve events controlled by the same edge may be called **conjugate events,** and it is important to note that changing the lap affects in opposite manner the two conjugate events which the edge controls.

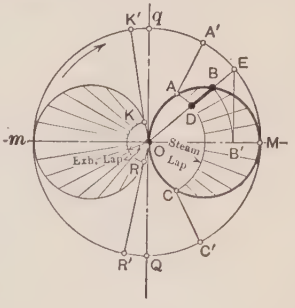


Fig. 135.

(e) Fig. 135 shows a **polar diagram of valve displacements,** corresponding to the different eccentric positions. This diagram is not necessary here, but it will be of use in connection with a valve diagram which will be discussed later. Given any eccentric position OE, the valve displacement $x = OB'$ is laid off as OB along OE. The locus

of point B is $OAMCORmK$, which is composed of two circles. The locus for displacements to the right is shown by the heavy line, and that for displacements to the left by the light line. Given any eccentric position such as OE, the intercept OB is the valve displacement (here to the right).

Arc AC is the *steam-lap line*, and is struck with the steam lap as radius. Arc RK is the *exhaust-lap line*, with radius equal to the exhaust lap.

When the eccentric is at Oq, the valve is central; when it extends through OA, the valve displacement equals the steam lap, and admission occurs; when at OM, the displacement and steam opening are maximum; when through OC, cut-off occurs; at OQ, the valve is central; when through OR, release takes place; at Om, the displacement and exhaust opening are maximum; and at OK, compression begins. The lengths of the radial section lines show the amounts of opening of the steam and exhaust edges.

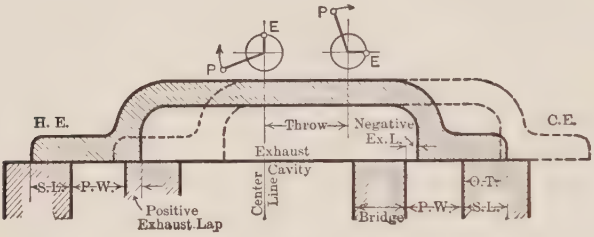


Fig. 136.

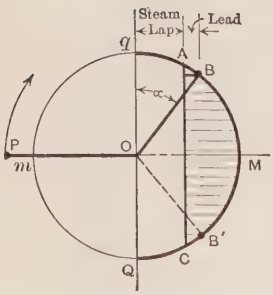


Fig. 137.

(f) Fig. 136 shows a double-end D-valve such as is used on double-acting engines. The crank-end displacement diagram will be similar to the head-end diagram rotated through 180 degrees.

145. Relative Valve and Piston Positions. (a) Consider the piston driven by a Scotch yoke, as in Fig. 132. When the crank is on head-end dead center, as in Fig. 137, the valve should have opened a slight amount, called the **lead,** principally in order to furnish steam to fill the clearance space and replace the vapor lost by initial condensation before the stroke starts. Hence

the eccentric at this time must be at OB, with displacement equal to *lap plus lead*. The angle $\alpha = qOB$ is called the **angle of advance,** and it is seen that, for the valve to have lead and the

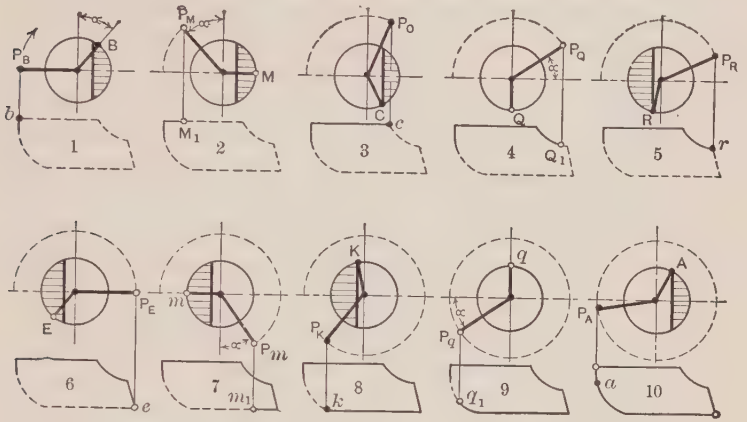


Fig. 138.

proper direction of motion when the crank is on dead center, the *eccentric must precede the crank by an angle equal to* 90 *degrees plus angle of advance.* Thus if rotation were counter-clockwise the eccentric would be at OB' when crank is at OP.

(b) Fig. 138 shows the successive critical crank positions during one complete revolution of the crank. These crank positions are located $90° + \alpha$ behind the corresponding eccentric positions. The figure also shows the development of the indicator diagram during the revolution.

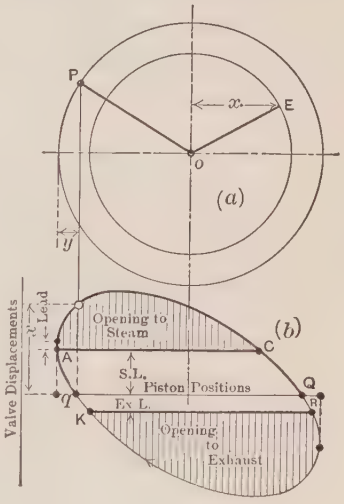


Fig. 139. — Elliptical Diagram.

146. Elliptical Diagram. To show at a glance the simultaneous displacements of the valve and piston throughout the complete revolution of the engine, the displacements of the valve may be plotted as ordinates on the corresponding positions of the piston as abscissas. These coördinates can be

obtained directly from the corresponding crank-pin and eccentric positions, as shown in Fig. 139 (*a*), or may be obtained from other valve diagrams which will be discussed later. The resulting figure is an ellipse, as shown in Fig. 139 (*b*), in which valve displacements to the right are positive ordinates and those to the left negative.

Lines *AC* and *RK* are the head-end steam and exhaust lines and are drawn at distances from *qQ* equal to these respective laps. (If the exhaust lap is negative, *RK* will be above *qQ*.) The valve events are lettered in accordance with the notation adopted in Section **143** (**k**). The valve openings are shown by the lengths of the section lines. The opening at the beginning of the stroke is the lead.

If an indicator diagram for the head end were drawn just below Fig. 139, the piston positions for the valve events could be found by vertical projection.

As both ends of the valve have the same displacement, the same ellipse would be used for the crank end, but the steam lap would be located below and the exhaust lap above *qQ* (if positive).

The elliptical diagram shows at a glance the complete action of the valve, and shows how the valve opening varies with the piston positions. The part of the diagram above the steam-lap line may be considered as a **Diagram of Steam Openings.** Similarly, that part lying below *RK* is a **Diagram of Exhaust Openings.**

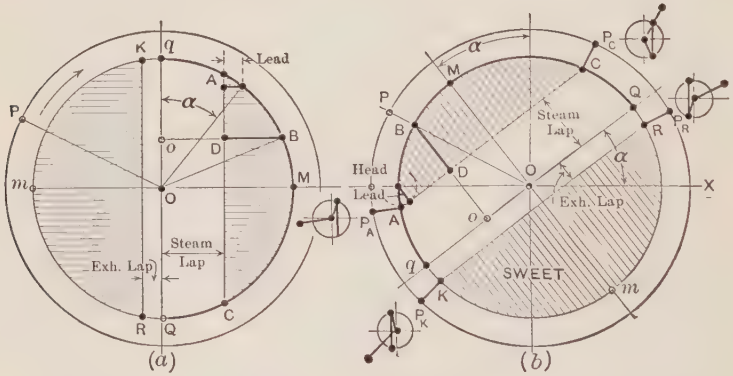


Fig. 140. — Sweet Diagram.

147. The Sweet Diagram. In Fig. 140 (*a*) is shown a diagram of valve and eccentric displacements similar to Fig. 134, and

Fig. 140 (*b*) shows the diagram rotated backward (counter-clockwise) through an angle of $90° + \alpha$, which is the angle at which the crank follows the eccentric. When the crank is at any location OP in Fig. 140 (*a*) and eccentric at corresponding position OB, the valve displacement is oB and its opening is DB. In Fig. 140 (*b*), with crank in same position OP, the distance oB, measured perpendicularly to qQ, gives the valve displacement, and DB is the valve opening. Thus the valve displacement and opening for any crank position can be obtained directly from Fig. 140 (*b*), which is called the "Sweet Diagram," and there is no necessity of finding the corresponding eccentric position.

In constructing the Sweet diagram a circle is drawn with radius equal to the eccentric throw; the axis qQ is at angle α with OX; and lap lines AC and RK are drawn at distances from qQ equal to the laps. OP_A is the crank position for admission; OM, for maximum displacement; OP_C, for cut-off; OQ, for valve central; OP_R, for release; OP_K, for compression. The openings are shown by the lengths of the section lines, and when the crank is on head dead center the opening is the lead. The foregoing is for the head end of a valve having positive exhaust lap. If the exhaust lap is negative RK would be above qQ. For the crank end of the valve the steam lap would be located below qQ, and the exhaust lap, if positive, above that line. The little "Pilot Diagrams" show the relation of crank and eccentric for all valve events.

The elliptical diagram can be obtained from Fig. 140 (*b*) by using distances oB as ordinates on the horizontal projection of the crank pin P.

148. Zeuner Diagram. In Fig. 141 (*a*) is shown a polar diagram of valve and eccentric displacements similar to Fig. 135, and Fig. 141 (*b*) shows the same diagram rotated backward through the angle $90° + \alpha$. In Fig. 141 (*a*), when the crank is at OP and eccentric at OE, the valve displacement is OB and its opening is DB. In the Zeuner diagram, Fig. 141 (*b*), with crank in the same position OP, OB is the displacement and DB is the opening. The crank positions for all events (major and minor) are shown and lettered on the figure, and the lengths of the radial section lines show the valve openings for the different crank positions. The lead is the opening when the crank is on

dead center. The Zeuner diagram for the crank end is similar to that for the head end rotated through 180 degrees. In Fig.

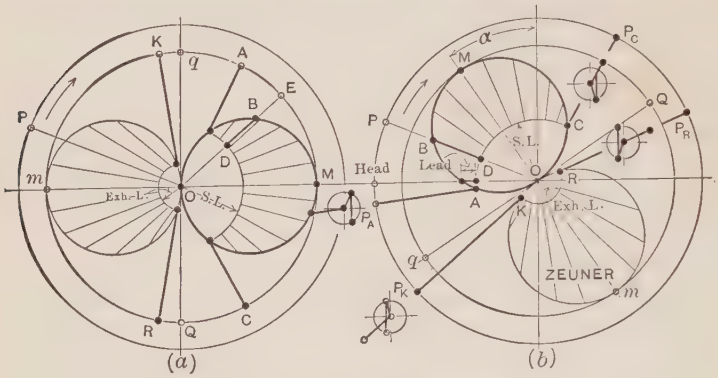


Fig. 141.—Zeuner Diagram.

141 (*b*) the "Pilot Diagrams" show the relative positions of crank and eccentric at all valve events.

The elliptical diagram can be easily obtained from the Zeuner diagram.

149. Bilgram Diagram. (a) The foregoing diagrams are useful for *analyzing* the action of a valve when its dimensions and those of the eccentric are known. They are difficult to use, however, in *designing* a new valve gear, that is, in determining the valve laps and the eccentric throw and angle of advance, which will give a proposed steam distribution in the cylinder, with specified widths of openings. The Bilgram diagram has the advantage that it can be readily used either for analysis or for design.

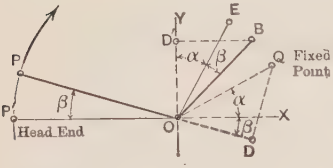


Fig. 142.—Bilgram Diagram.

(b) Fig. 142 shows the principle of the Bilgram diagram. On it the line OQ is made equal to the eccentric throw and is at angle α (the angle of advance) with OX. Q is a fixed point on this diagram and is called the **Lap-Circle Center.** The **Fundamental Principle** on which the construction and use of the Bilgram diagram is based may be stated thus: *The length of the perpendicular (QD in Fig. 142) from the lap-circle center (Q) to the crank (OP), produced if necessary, is the valve displacement corresponding*

to that crank position. Proof. — When the crank is on dead center (at OP' in Fig. 142) the eccentric is at OE and angle $EOY = \alpha$. Now if the crank rotates through angle β to OP, the eccentric moves through the same angle to B, and the valve then has displacement equal to $D'B$. Now angle $QOX = \alpha$ and $XOD = \beta$. Then if QD is drawn perpendicular to OP (produced), it is evident that triangles OQD and OBD' are equal and that $QD = D'B$; hence the perpendicular QD gives the valve displacement when crank is at OP, which proves the "fundamental principle."

The term "perpendicular" used in connection with the Bilgram diagram will hereafter be understood to refer to the length of perpendicular dropped from Q to the crank, produced if necessary.

The elliptical diagram can of course be constructed by using these perpendiculars as ordinates on piston positions as abscissas.

(c) Evidently the feet of the perpendiculars will be on a circle with OQ as diameter, as in Fig. 143. By subtracting the lap from the displacement perpendiculars, the openings of the valve are obtained.

In Fig. 143, with Q as center and radius equal to the steam lap, the *steam-lap circle* BF is drawn; hence the lengths of

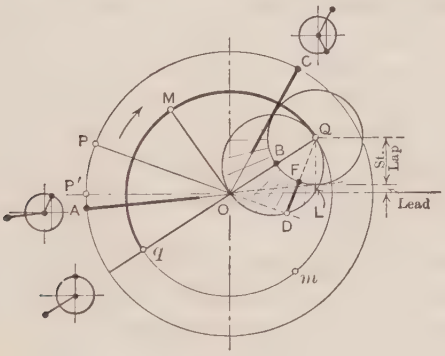


Fig. 143.

the section lines (drawn radially from Q) in this figure give the steam openings for the head end of the valve. Then OA (whose extension is tangent to the lap circle) is the crank position for admission, as the valve displacement (as shown by the length of perpendicular) just equals the steam lap. At OP the opening

is FD and displacement is QD; at OM the opening is maximum and equal to OB; at OC the opening is zero and cut-off occurs. When the crank is on head-end dead center the opening L is the lead.

(d) Fig. 144 shows the completed Bilgram diagram for the head end of the valve. Compared with Fig. 143, it is seen that the smaller circle about Q, and the crank positions for the exhaust events, have been added. The small circle is the **Exhaust-Lap Circle,** with radius equal to the exhaust lap. If **release** is to occur when the crank is in position OR, the exhaust-lap circle must be tangent to this line, for then the valve displacement (as shown by the length of the perpendicular) is equal to the exhaust lap.

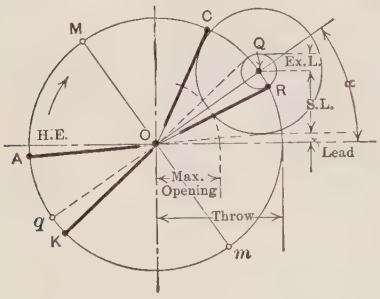


Fig. 144.

When the crank coincided with OQ the valve was central, and, since in this case the valve does not open until the crank has rotated clockwise past OQ, the valve is closed when central, therefore the exhaust lap is positive.

(e) The portions of the perpendiculars beyond the exhaust-lap circle represent exhaust openings. Evidently exhaust closure, or *compression*, takes place when the extension of the crank is tangent to the upper side of the lap circle; thus OK is the crank position for compression. When the crank coincides with Oq the valve is central; and, since in this case the exhaust lap is positive, the exhaust closure must take place before the valve reaches central position; hence OK is below Oq in this case, as the rotation is clockwise.

If the exhaust edge has **negative lap,** the crank position OR would be tangent to the upper side of the exhaust-lap circle, and the extension of OK would be tangent to the under side.

(f) The **application to design** problems when certain definite cut-off, lead opening, and maximum valve opening to steam are required, is as follows: In Fig. 145, for the H.E. of the valve, starting with the X and Y axes, draw the desired crank position OC for cut-off; draw a line (L) parallel to OX and above it at a distance equal to the specified lead; and with O as center and radius equal to the desired maximum valve opening strike an

arc B in the position shown. From what has gone before, it is evident that the steam-lap circle must be tangent to these three lines. The location of its center Q can usually be found as quickly and as accurately by trial as by geometrical construction. Having the point Q determined and the steam-lap circle drawn, the diagram then shows the steam lap and the throw and angle of advance of the eccentric, which must be used to obtain the desired results.

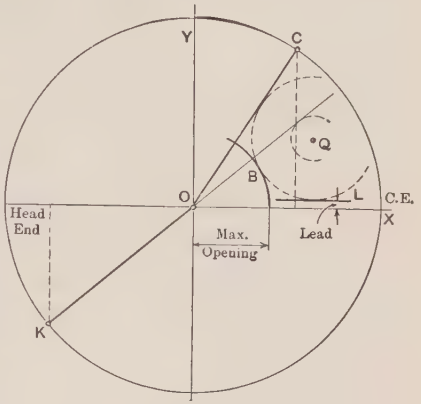


Fig. 145.

If OK in Fig. 145 is the desired crank position for compression, the exhaust-lap circle would be drawn tangent to the extension of this line, with center at Q just found, and its radius will equal the exhaust lap. Whether the exhaust lap is positive or negative can be determined in accordance with (e) in the foregoing discussion.

(g) For the **crank end** of the valve, the Bilgram diagram would be similarly constructed but rotated 180 degrees with respect to the diagram for the head end. Q and q must of course be diametrically opposite each other. Fig. 146 shows the crank-end Bilgram diagram separately; * it is usually, however, drawn superimposed on the diagram of the head end.

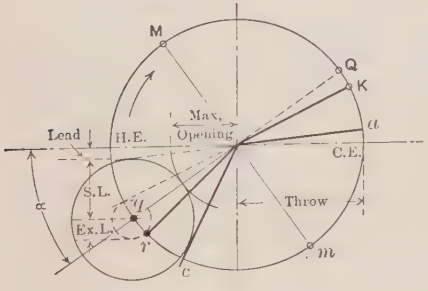


Fig. 146.

150. Distortion Due to Angularity of the Connecting Rod. In Fig. 147 o is the middle of the stroke and the distance oO is equal to the length of the rod aP. If an infinite rod is used, the displacement of the piston oa will of course be equal to the displace-

* With negative exhaust lap (shown dotted).

ment of the pin P, which is equal to OA. If, however, a finite rod is used these displacements will not be equal. For, if the end of the rod a is kept stationary and the other end P is uncoupled and swung to A', then oa will be equal to OA', which is seen to be greater than OA. It will be found that no matter where the crank is, A' will always be to the right of A. It is evident that, owing to the

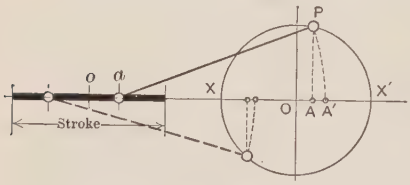


Fig. 147.

"angularity" of the connecting rod, if one of finite length is used, *the piston is always nearer the crank end of the stroke* than it would be ideally, except of course when it is at the end of its stroke.

It follows that: *The valve events occur later with respect to piston positions during the forward stroke and earlier in the return stroke* than they would with the Scotch yoke, but their mean is the same as this latter gives if the laps are equal.

The distance AA' is the "distortion due to the angularity of the rod" and is equal to the difference between the length of the rod and its horizontal projection. This distortion is greatest when the crank is at right angles to the center line of the engine, and decreases to zero at the ends of the stroke. The shorter the length of the rod when compared to the crank radius, the greater is this relative distortion.

If the diameter of the crank circle XX' represents the stroke of the piston, then, having any position, such as A', the corresponding position of the crank pin P may of course be found by drawing the "connecting-rod arc" $A'P$; or if P is known at the start, A' may be found from it in a similar manner.

The **angularity of the eccentric rod** can be neglected in most cases, as the rod is usually very long when compared with the eccentric throw.

151. Valve Diagrams Considering "Angularity" of the Connecting Rod. (a) All the valve diagrams discussed show the true positions of the *crank;* therefore if the positions of the piston are not being considered, but only those of the crank, the angularity of the connecting rod would not affect the diagram.

If, however, after the crank positions have been found, the true positions of the piston are desired, it will then be necessary to consider the angularity. Having already determined the crank positions, the corresponding true position of the piston would be found by drawing the connecting-rod arcs in the manner shown in Fig. 148 (*a*), (*b*), (*c*) for the Sweet, Zeuner, and Bilgram diagrams. Should the piston positions be known at the outset, then by drawing similar arcs the true crank positions can be found, and these would be used in constructing the rest of the diagram.

In the elliptical diagram, Fig. 148 (*d*), it is evident that the angularity causes all points on the ellipse to be displaced toward

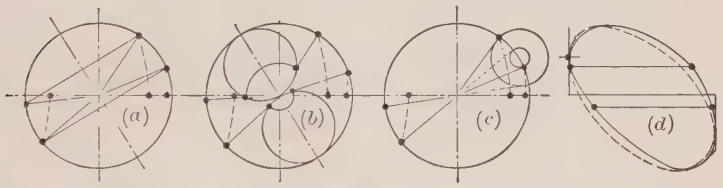


Fig. 148.

the crank end of the stroke. The resulting figure is of oval shape, in consequence of which the diagram is sometimes called the "Oval Diagram."

(b) Owing to the effect of the angularity of the connecting rod, the piston displacement for similar events in the two strokes of a double-acting engine will not be equal if the laps on the two ends of the valves are the same. It is possible to "equalize" the cut-offs by using unequal steam laps, but in that case the conjugate events (admissions) are unequal. Similarly, the compressions can be "equalized" by using unequal exhaust laps, but then the releases will be unequal.* Equalization may also be accomplished by using special arrangements of rockers between eccentric rod and valve rod. This matter is discussed fully in most books especially devoted to valve gears, and will not be considered further here.

152. Valve and Port Openings. For the rate at which the steam is supplied to the cylinder to be always equal to the rate

* Except in the special case in which the exhaust lap would be zero if the "angularity" of the connecting rod were neglected.

at which volume is made available by the piston, the following expression must be satisfied:

$$av = AV, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (273)$$

in which

a = area of passage (sq. in., usually);
A = area of piston (same unit);
v = velocity of steam (ft./min., usually);
V = velocity of piston (same unit).

Then

$$a = AV \div v. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (274)$$

Valves are usually designed to have a maximum area of opening which corresponds to a velocity (v) of steam which has been found by experience to give satisfactory results. The **maximum valve opening** (a) is computed by using Eq. (274), in which V is the mean piston velocity (equal to 2 × stroke in feet × r.p.m.), and v has a value which in practice varies from 6000 to 10,000 feet per minute, but is usually about 8000 feet per minute in simple engines.

In designing the gear with single valve, it is generally only necessary to see that the steam opening of the valve is sufficiently large, for the exhaust opening will always be more than is required because the exhaust lap is very much smaller than the steam lap.

The **width of the valve opening** (used in constructing the valve diagrams) is of course equal to the area of opening divided by its length. This length is nearly always equal to the length of port across the cylinder.

In case a simple valve is used, the **area of the port** in the cylinder should be sufficient for accommodating the exhaust steam, as the same passage is used for both the entering and the outgoing vapor. Its area may be determined from Eq. (274), using for the steam velocity (v) from 4500 to 7000 feet per minute, but about 6000 is usual in simple engines. This area is then more than is needed for the admission of steam.

If the exhaust passage is separate from the steam passage, this latter can have area about equal to the maximum valve opening to steam.

153. **Cushioning the Reciprocating Parts.** (a) First suppose there is no compression. Then when the piston approaches the end of its stroke the effective steam pressure and the inertia of

the reciprocating parts are both acting towards that end of the stroke, taking up the slack in the bearings of the reciprocating parts. Now, when the steam is admitted on the other side of the piston, the pressure on the bearings is reversed more or less suddenly. With reciprocating parts of small weight, and with high steam pressure, this reversal will be very sudden, and if there is much "play" in the bearings (and there must always be a little) the consequent impact or "hammering" will cause excessive stresses in the impinging parts, and will render the operation of the engine noisy.

(b) One method of preventing the occurrence of these undesirable features is to make the weight of the reciprocating parts so great that their inertia will oppose the pressure of the entering steam sufficiently to cause the play in the bearings to be taken up gradually, thus preventing impact. But as the inertia forces are free forces which tend to move the engine on its foundation, it is usually desirable to have them small, even when counterbalancing is attempted; so this method is usually unsatisfactory.

(c) Another method is to arrange the valve to open gradually, but this is accompanied by a more gradual cut-off, which is undesirable.

(d) The best method is to gradually reverse the pressure on the bearings by introducing compression; then, when admission takes place, there is no play to be taken up and consequently no impact.

It is possible to compute the inertia force of the reciprocating parts at the end of the stroke. Then in order to reverse the pressure on the bearings the steam pressure at the end of compression should equal or be greater than this inertia force plus the steam pressure on the other side of the piston.

154. Early Valve Opening. (a) If steam is admitted just as the new stroke begins, the pressure will not rise immediately to the value in the steam pipe because (1) the valve opens gradually, (2) the clearance space must be filled, and (3) a large proportion of the entering steam is liquefied by cylinder condensation. Hence it is necessary to have the valve open before the commencement of the stroke; that is, the valve is given **"lead."** As the clearance volume is constant, the lead and crank angle at which opening occurs should be constant regardless of variations

in cut-off, if the speed of the engine is uniform. The higher the speed and the less the compression, the earlier should the opening of the valve occur.

(b) In order to have the steam pressure drop to that of exhaust by the time the end of the stroke is reached, the exhaust edge of the valve is given lead, causing **"early release."** As release and compression are conjugate events, the fixing of one of these events determines the other. Often it is not possible to have both occur as desired, in which case a compromise must be made.

155. Limitations of the Simple Valve. It is impracticable to have cut-off occur early in the stroke with the simple D-valve because, in order to obtain a satisfactory width of opening in such cases, it is found (1) that the valve and eccentric are excessively large (and consequently the valve gear must work against great friction and inertia forces), and (2) that the release and compression occur too early in the stroke.

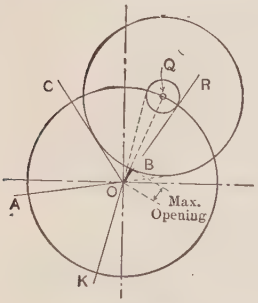


Fig. 149.

Fig. 149 is a Bilgram diagram for a valve cutting off at one-fourth stroke. If the scale is such that the maximum opening to steam is one inch, it is seen that all of the foregoing statements are true. The simple D-valve is not used with cut-offs much earlier than five-eighths stroke.

Ordinarily, the best economy in a simple engine is obtained when cut-off is at about one-fourth stroke; hence the simple slide valve should not be used when economy is important.

156. Special Types of Single Valves. (a) By increasing the length of the steam edge of the valve a reduction can be made in the port width, laps, travel, and eccentric throw; but there are practical limitations to increasing the length of this edge in the simple flat valve.

(b) **Piston Valves,** Fig. 150, which may be looked upon as flat valves rolled into cylindrical shape, may have greater length of edge (equal to the circumference) than the simple flat valve, without having prohibitive size. Fig. 150 (*a*) shows an "external" piston valve; Fig. 150 (*b*) shows an "internal" one.

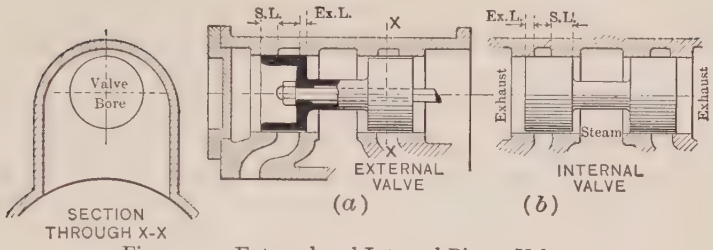


Fig. 150. — External and Internal Piston Valves.

(c) **Multiported Valves,** in which there are two or more working edges, are frequently used. Fig. 151 shows a **"Double-ported Marine Valve,"** each end of which has two steam edges and two exhaust edges.

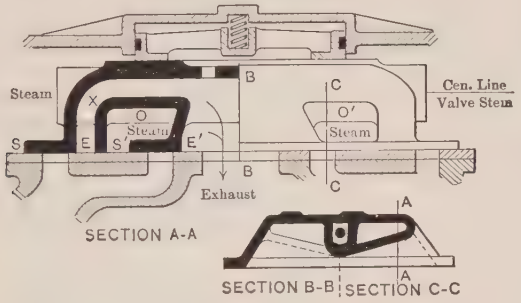


Fig. 151. — Double-ported Marine Valves.

(d) Some valves have **auxiliary ports** in them so arranged as to give multiported action. An example of this is the **Allen or Trick Valve** shown in Fig. 152. This valve has an auxiliary passage *aa'* and valve seat so arranged that, as the valve moves to the right in the figure, the edge *f* opens simultaneously with the main steam edge *y*. The exhaust is single-ported.

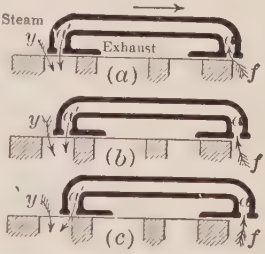


Fig. 152. — Allen Valve.

Considering the valve as moving to the right, the phases of opening of the steam edge are: (1) *"Double-ported"* action while edges *f* and *y* open at the same rate. This continues until the auxiliary port *a* is wide open. (2) With movement continuing, Fig. 152 (*b*), the opening at *y* increases but that through *a* remains constant,

i.e., the opening is "*single-ported plus a constant.*" This continues until a' in Fig. 152 (*c*) becomes throttled by the exhaust edge of the valve seat. (3) As the opening at a' is then decreasing (as the valve continues to the right) at the same rate as that at edge y is increasing, the effective area remains "*constant.*" (4) If the movement is sufficient to completely close a', the valve becomes "*single-ported.*" Now if the valve returns to the left to close, the effective openings will decrease in the reverse order.

The openings of the steam edge of an ordinary valve are shown by the sectioned part above the steam-lap line of the elliptical diagram in Fig. 139. This is also shown (somewhat distorted) by the light line in Fig. 153. The heavy lines in this figure show the character of the openings when an Allen valve, like that just described, is used. Note that the smaller openings are affected more than the larger ones.

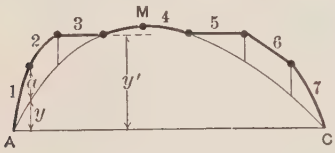


Fig. 153.

Piston valves similarly arranged have been used (Armington-Sims valve).

(e) The **Sweet Valve** shown in Fig. 154 is another valve having an auxiliary port. This valve is a rectangular piston valve which slides between the valve seat and a "**balance plate,**" which latter is supported by distance pieces so as to just clear the valve. The face of the balance plate is similar to the face of the valve seat. All sliding surfaces are scraped to give sufficient clearance for free movement of valve but not enough to permit of appreciable leakage of steam. The valve has separate auxiliary ports a at each end, which causes double-ported action through at least part of the opening of the valve.

Referring to Fig. 155, it is seen that as the valve moves to the right the phases of opening of the steam edge are the same as those of the Allen valve; thus there is (1) "*double opening,*" with y and f opening together; (2) "*single-ported plus a constant,*" when f becomes greater than a; (3) "*constant,*" when a is being closed by the exhaust edge of the valve seat at same rate that y is opening; and (4) "*single-ported,*" when a is entirely closed by

the exhaust edge. The areas during closure decrease in the reverse order.

The auxiliary port, or another one, may be so arranged as to assist during the exhaust.

(f) A combination of the Allen and Sweet arrangements gives quadruple openings (Woodbury valve), and there are many

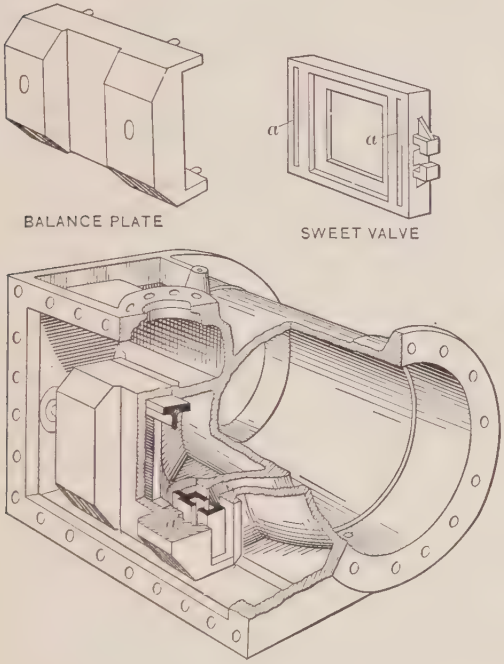


Fig. 154. — Cylinder with Sweet Valve.

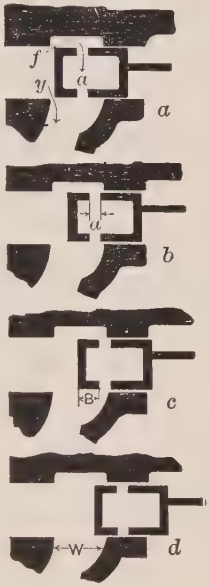


Fig. 155.

other forms of such valves. For further discussion see text-books on Valve Gears.

(g) **Valve Friction** is undesirable, not only because of the waste of power it causes, but because it may disturb the action of the governor, if one is used. The whole back of the simple D-valve is subject to full steam pressure, while the larger part of the under side is exposed to the exhaust pressure. The unbalanced pressure causes excessive wear and friction loss at the rubbing surfaces. To reduce this unbalanced pressure, various schemes of "**balancing**" are used. The simplest is the use of

a piston valve which is perfectly balanced except for its weight. The Sweet type of valve is practically the equivalent of the piston valve in this respect. Some valves have **"balance or equilibrium rings"** on their backs (like that shown in Fig. 156) so arranged that the area within the ring is subject to exhaust pressure and is about equal to the area subjected to exhaust pressure on the under side. However, in such cases there should always be enough unbalanced pressure to maintain steam-tightness between valve and seat.

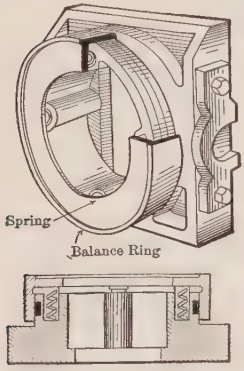


Fig. 156. — Valve with Balance Ring.

(**h**) In case there should be entrapped in the cylinder a quantity of water more than sufficient to fill the clearance space, some part of the engine would break during compression unless some means of relief were provided. Sometimes **"Relief Valves,"** which are somewhat similar to boiler safety valves, are attached to the cylinder ends. Sometimes the slide valve itself offers this relief; for example, the simple slide valve, with or without balance rings, and valves (like the Sweet) with pressure plates can lift from their seats and thus give relief. In such cases there should be springs or other devices to return the valve and balance ring, or plate, to its proper position after the cylinder has been relieved of the water. Piston valves offer no such relief themselves, and engines with this type of valve should be provided with special relief valves or other similar device.

157. Valve Gears for High-Speed Engines. (**a**) The high-speed engine was briefly described on page 245. It is fitted with a "shaft governor," which controls the point of cut-off by varying the position of the eccentric with respect to the crank.

(**b**) Simple engines of this type usually have cut-off occur at about one-fourth stroke when operating under "normal load." The "*range of cut-off*" is generally from zero to five-eighths or three-fourths stroke, and the "*range of load*" is from "friction load" to 50 or even 100 per cent overload.

It was shown on page 288 that if a simple slide valve is used to give cut-off as early as one-fourth stroke, certain features will

be introduced which are undesirable in the ordinary case. Two of these are early release and early compression.

(c) It so happens, however, that these phenomena are desirable in a high-speed engine. The early release is advantageous, as it allows more perfect drop to back pressure in the short time available. The early compression assists in causing a gradual absorption of the inertia forces; and an excessive terminal pressure can be prevented by increasing the ratio between clearance volume and piston displacement. With a given clearance volume and piston area, this ratio can be made large by using a short stroke, and with high rotative speed a short stroke is desirable in order to keep the piston speed down to safe limits; hence what are faults in the ordinary case make the short-stroke high-speed engine possible.

(d) The *excessive size of the valve-gear parts*, which ordinarily occurs when a valve is designed for an early cut-off, as was shown in Section 155, may be overcome by increasing the length of port, which calls for a narrower valve opening and a corresponding reduction of the laps, travel, and size of the eccentric; and these in turn are accompanied by a decrease in the friction and wear of the valve gear. The greater length of port may be obtained by using a wide valve, a piston valve, or a multiported valve.

With the type of valve gear which is used on this class of engine, the travel of the valve varies with the cut-offs, and the earlier the cut-offs the more restricted are the openings of the valve. The valves may be designed to have openings ample for the latest cut-offs, and to have auxiliary ports added in such a manner as to assist during the early openings only, and to have little or no effect on the wider openings. Examples of these various types of valves have already been given.

Sometimes *special arrangements* of linkage are employed to give wide openings with small travel, as in the case of the "High-Speed Corliss" engine to be considered later.

(e) The friction of the valve is undesirable, not only because it decreases the mechanical efficiency of the engine and causes wear, but also because it disturbs the action of the shaft governor. This latter is especially true if the governor is of the Inertia type. The governor is also affected by the inertia of

the valve gear; hence high-speed engines use valves that are balanced and are of light weight.

(f) In Fig. 157, which is a diagram of positions, *1–3* is one path, with respect to crank *OP*, over which the eccentric might be shifted by a shaft governor in adjusting the cut-off to meet the power demanded of the engine. When, in Fig. 157 (*a*), the eccentric is at *1* (with throw *O–1* and angle of advance α_1) the cut-off is at about three-fourths stroke, for when the eccentric has rotated in the direction of the arrow to c_1 (displacement

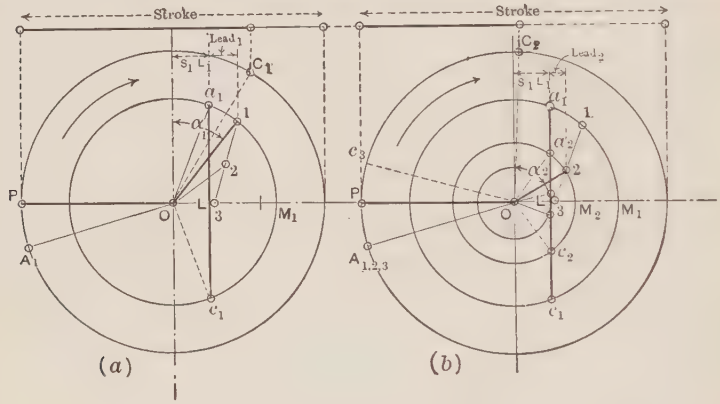


Fig. 157. — Diagram of Positions.

equal to steam lap) the crank pin is at C_1; when, in Fig. 157 (*b*), the eccentric is at *2* (with throw *O–2* and angle of advance α_2) the cut-off is about one-half stroke, for when the eccentric is at c_2 the crank pin is at C_2; and when the eccentric is at *3*, diametrically opposite the crank, the cut-off is at C_3. In this case the path *1–3* is so selected that it will coincide with La_1 when the crank is in position OA_1; hence the crank-pin positions (A_1, A_2, and A_3) for the admissions corresponding to all cut-offs will coincide, that is, the admission is constant.

With crank at *OP*, the horizontal distance between the eccentric and the steam-lap line a_1c_1 is the lead. From the figure it is seen that Lead_2 is less than Lead_1, and that as the eccentric is shifted to give earlier cut-off the lead becomes less.

The figure also shows that when the eccentric throw is *O–1* the maximum valve opening is LM_1, when the throw is *O–2* the maximum opening is LM_2, and with *O–3* this opening is *L–3*. Thus the maximum openings decrease as the cut-off is made to

occur earlier. The valves are therefore usually designed to have proper opening at latest cut-off when operating as a single-ported valve and to be multiported when early cut-offs occur.

If the exhaust-lap line is added to the diagram of positions, Fig. 157, and the crank positions are determined for exhaust events corresponding to the different points of cut-off, it will be found that as cut-off is made to occur earlier the release and compression are also made earlier. Thus with early cut-off

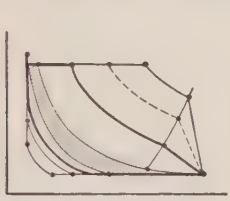


Fig. 158.

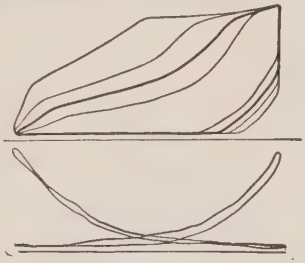


Fig. 159.

there is greater compression than with late cut-off. Fig. 158 shows how these events vary and affect the form of the ideal indicator diagram, and Fig. 159 is an actual diagram obtained from a high-speed engine.

From the foregoing it is seen that the following general statement can be made: *As the eccentric is shifted from the outer to the inner position, cut-off, release, and compression are made to occur earlier and the maximum opening is decreased.*

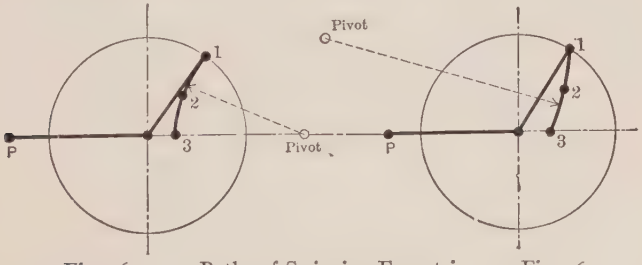


Fig. 160. Paths of Swinging Eccentric. Fig. 161.

Instead of shifting the eccentric over a straight path, it can be swung about a pivot in the governor case or flywheel. When an inertia governor (Fig. 130) is used with external valve, the eccentric center will be moved over a circular path, as in Fig. 160, with

pivot opposite the crank. When the ordinary centrifugal governor is used, the eccentric may be pivoted on either side of the shaft with respect to crank, hence its path may be that in Fig. 160 or that in Fig. 161. These curved paths approximate the straight one in Fig. 157. The admission will vary as the eccentric position is changed, and the character of the variation depends on the curvature of the path.

(g) The various valve-gear diagrams can be constructed in the usual manner, taking each position of the eccentric inde-

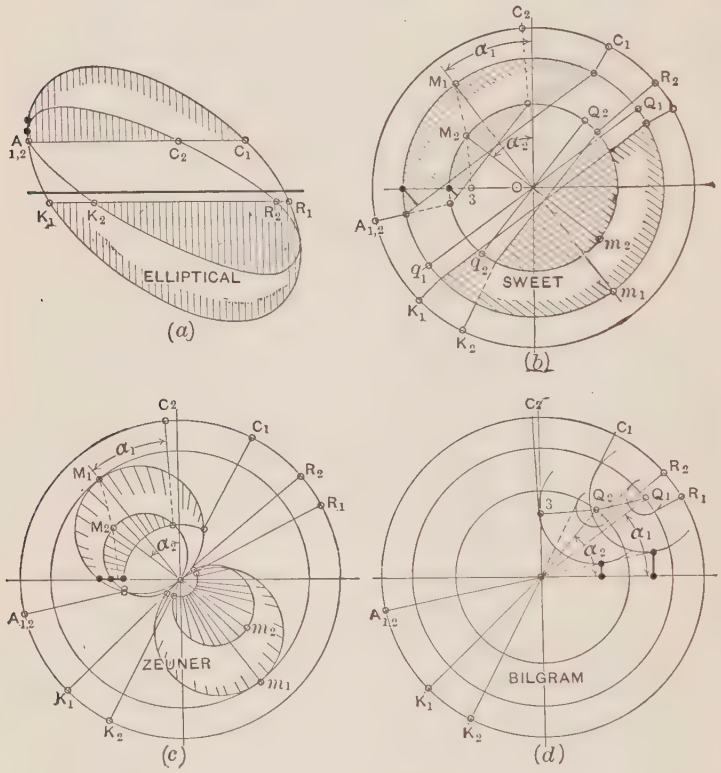


Fig. 162. — Valve Diagrams for Variable Eccentric Valve Gear.

pendently. The diagrams for the different eccentric positions may be drawn separately, but usually they are superimposed on one another, as in Fig. 162, which corresponds to Fig. 160.

(h) At any one cut-off the valve events can be equalized in the manner mentioned in Section 151 (b). In some cases it is

possible to approximate the equalization in all positions of the eccentric by using a special arrangement of rocker arm or guide for the eccentric-rod pin.

158. General Characteristics of Independent Cut-off Gears. (a) If an engine operates at a constant speed, as most engines do, it is desirable to have the admission, compression, and release remain constant, no matter how the cut-off varies. It has been seen that the simple valve with shifting eccentric does not give this desired constancy of all these events, nor does it give sufficient opening and sharp closure when cut-off is early in the stroke. If, however, instead of a single valve, two or more are used, with two independent sets of valve gear, it is possible to avoid some or all of these difficulties. Owing to the complications and extra expense involved with such arrangements, they are not often used on the small high-speed types of engines. They are quite common on larger and longer stroke, medium speed engines.

(b) In general the gear having the two valves may be arranged in either of two ways: (1) Each valve may control one pair of conjugate events; thus, one valve, having a fixed eccentric, may operate the release and compression, while the other valve takes care of the admission and cut-off, the latter event being changed by shifting the eccentric by a shaft governor, as in the case of the valve gear used on high-speed engines. In this case, separate valve diagrams would be drawn for each pair of conjugate events; that for the steam events would be constructed the same as for the simple-shifting eccentric gear; that for the exhaust events would be constructed the same as for any case in which the crank positions for the opening and closing of the exhaust edge are given, together with the maximum width of opening desired. This arrangement will not be considered further. (2) In the other arrangement of the gear, one valve, which will be termed the *main valve*, controls the admission, compression, and release, and is driven by a fixed eccentric ("main eccentric"). The other valve operates the cut-off only, and will be called the *cut-off valve*. It is an intercepting valve, being located between the main valve and the source of steam supply. It may slide on a separate valve seat, or it may ride on the back of the main valve, in which case it is called a "*riding*" *cut-off valve.*

The variation in cut-off may be accomplished in three ways: (*a*) by changing the lap of the cut-off valve, (*b*) by changing the position of the cut-off eccentric with respect to the crank, and (*c*) by a combination of (*a*) and (*b*).

The range of cut-off on medium-speed engines is usually from zero to five-eighths or three-fourths stroke. Under normal load simple engines usually cut-off at about one-fourth stroke.

(c) In *all* the gears having a cut-off valve of the intercepting type, the **main valve** or valves (operated by the main eccentric) control the admission, release, and compression.

Referring to Fig. 144 it is seen that the line OQ bisects the angle formed between OR and OK produced; hence, in order to determine the proportions of the main valve and its eccentric, proceed as follows:

The two conjugate events (R and K), which can be decided upon initially, fix the angle of advance of the main eccentric, for, in the Bilgram diagram, Fig. 163, OQ_1 must bisect the angle

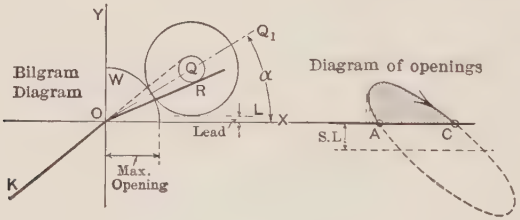


Fig. 163.

Fig. 164.

between R and K (produced). After drawing the lead line L, and the arc W for maximum opening, the steam lap is determined by drawing the lap circle tangent to W and L in the figure, the center, Q, being on OQ_1. Having located Q, the exhaust-lap circles can then be drawn. The diagram is now complete and all dimensions for the valve and eccentric have been found. The cut-off of the main valve is unimportant, provided it is at least as great as the latest given by the cut-off valve. Compression, of course, can be equalized by using unequal exhaust laps.

The diagram of openings of the main valve, which is that part of the elliptical diagram which lies above the steam lap, *S.L.*, (for the H.E. of the valve), is shown in Fig. 164.

(d) In *all* valve gears in which there is a separate cut-off

valve it is necessary that this valve open before the main valve does, as the latter controls the admission. To provide for this, it is necessary in many instances to have negative lap on the cut-off valve and large angle of advance, α', of the cut-off eccentric (α' may even be greater than 180 degrees in some instances).

In the general case the constructions of the various valve diagrams for negative lap and large angle of advance are identical

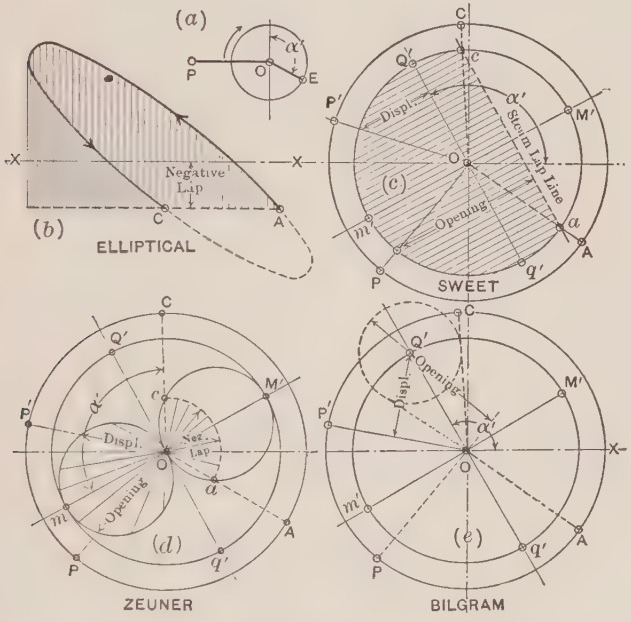


Fig. 165.

with those previously described; in each case the angle of advance is located in exactly the same manner, and the negative lap is laid off opposite the positive lap; the openings are then equal to the displacement plus the negative lap, and the closures equal the displacement minus this lap. With negative lap the valve when central is open, hence closure must occur after the central position has been passed, and opening takes place before that position is reached. Figs. 165 (*b*) to (*e*) show the various valve diagrams for angle of advance greater than 90 degrees, and Fig. 165 (*a*) gives the actual position of the eccentric with

respect to the crank when the latter is on dead center. The lengths of the section lines show the widths of valve openings.

The generating point in the elliptical diagram will move around the ellipse in a counter-clockwise direction if the angle of advance is greater than 90 degrees, as it is in Fig. 165. As before, the part of the elliptical diagram lying above the steam-lap line constitutes a *diagram of openings*.

The cut-offs at one point in the stroke can be equalized by using unequal laps.

(e) Referring to Fig. 165 (*e*), it is seen that the cut-off can be varied either (1) by altering the size of the lap circle (which may even be made positive), or (2) by changing the angle of advance, α'. Both methods are used in practice.

A shaft governor may be used to automatically change the angle of advance by turning the cut-off eccentric about the center of the shaft on which it is loosely mounted.

159. Independent Cut-off Valve with Stationary Seat. Fig. 166 shows diagrammatically an arrangement with cut-off valve *C* riding on an independent valve seat between the main valve *M* and the steam pipe. The main valve is driven by a fixed eccentric. It controls the admission, release, and compression, and is designed in the manner outlined in Section 158 (c). The cut-off valve is driven by an independent eccentric and controls only the one event. The cut-off can be changed in either of two ways already mentioned.

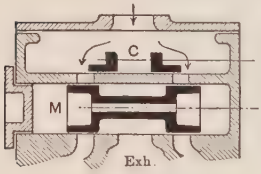


Fig. 166.

Case I. — The lap of the cut-off valve may be altered by any of the following methods:

(1) The valve may be in two parts, mounted on the valve stem with R.H. and L.H. threads respectively. By turning the stem, the distance between the ends, and consequently the laps, can be varied. With this arrangement, as in Fig. 167, the adjustment is made by hand, and the point of cut-off for the setting can be read on the indicator, which is moved by a nonrotating nut on the valve stem. It is difficult to arrange a governor to make the adjustment with this arrangement, as several revolutions of the valve stem are required to accomplish the full range of cut-off.

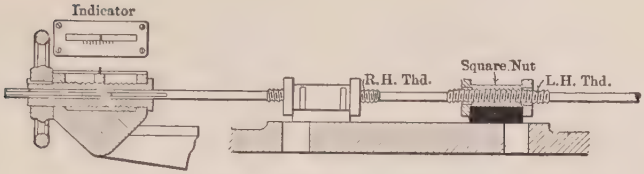


Fig. 167.

(2) Fig. 168 shows the back of another arrangement in which the edges of the valve and ports are oblique. On the back of the valve is a rack with which a pinion, on the valve stem, engages. By turning the stem the valve may be raised or lowered (as viewed in the figure), thus changing the distance between its edge and that of the port.

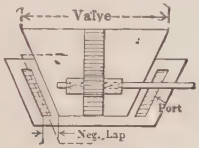


Fig. 168.

Fig. 169.

(3) Fig. 169 shows a somewhat similar arrangement, except that the valve face and seat are cylindrical surfaces. The valve is fastened to the stem, so that by turning the latter the lap is changed.

The arrangements shown in Figs. 168 and 169 can be controlled by a fly-ball governor, which can be connected to an arm on the valve stem.

Case II. — The angle of advance may be changed, as in Fig. 170, in which M is the main eccentric, 0, $\frac{1}{4}$ and $\frac{3}{4}$ are the positions of the cut-off eccentric for those cut-offs, λ is the **"range angle"** through which the governor has to turn the eccentric on the shaft, α_0' is the maximum angle of advance of the cut-off eccentric, and $\alpha_{\frac{3}{4}}'$ is that for three-fourths cut-off.

Fig. 171 shows for this case the diagram of openings of the cut-off valve (dotted lines) superimposed on that for the main valve (heavy lines), and the sectioning shows the effective opening from the time of admission of the main valve to the closure of the cut-off valve at one-fourth stroke.

The arrangement of valves shown in Fig. 166 is not satisfactory, as with early cut-offs the space beyond the end of the main

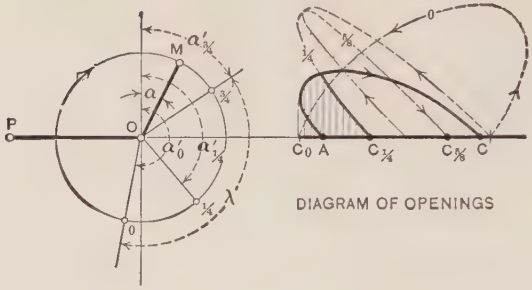


Fig. 170. Fig. 171.

valve is clearance space during the part of the expansion preceding the closure of the main valve.

160. Riding Cut-off Valves. (a) Instead of having a separate seat for the cut-off valve, this valve may ride directly on the back of the main valve (or within it, if piston valves are used) and perform its functions with respect to a port in that valve. There are several such arrangements possible. One, the **Buckeye Gear**, is in effect the exact equivalent of the arrangement described as Case II above.

The arrangement of valves in this gear is given in Fig. 172, both valves being shown central with respect to the ports. The

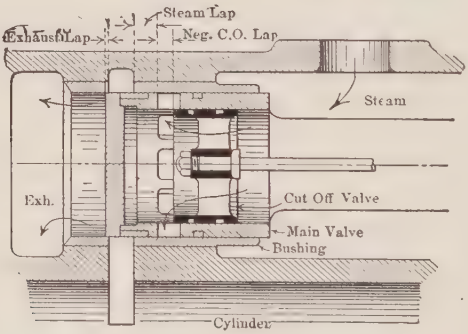


Fig. 172.

cut-off valve has negative lap equal to the amount it is open in the figure with respect to the main valve. The main valve is a box filled with live steam, practically a reciprocating steam chest. It is an " internal " valve, taking steam from the inside and exhausting at the ends. The cut-off valve rides inside the

main valve and is "external"; its negative lap is of constant amount. Its valve stem passes through that for the main valve, the latter being hollow.

The main valve is driven by a fixed eccentric, and controls admission, release, and compression. The cut-off valve is driven by an eccentric which is controlled by a shaft governor which turns the eccentric about the center of the shaft, thus varying the angle of advance, α', as in Fig. 170.

(b) The arrangement of the rockers which guide the eccentric-rod ends is shown in Fig. 173. The main rocker ab is pivoted at b

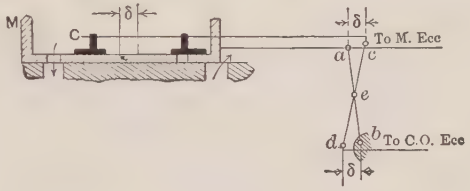


Fig. 173.

to the frame of the engine. The cut-off rocker cd is pivoted at its middle e to the middle of the main rocker. With this special arrangement of rockers, it is seen that the displacement (δ) of the center of the cut-off valve with respect to the center of the main valve is given by the distance between a and c, which is the same as that between b and d. Evidently, then, the motion of the cut-off valve with respect to the main valve is the same as that of c with respect to a, or opposite to that of d with respect to b. Since b is a *fixed point*, it follows that *the motion of the cut-off valve with respect to the main valve is the same as that of a simple valve with respect to a fixed seat.* The distance the cut-off valve travels with respect to the main valve remains constant, no matter how the cut-off and phase relations of the two valves are altered by changing the angle of advance of the cut-off eccentric.

Thus this arrangement is equivalent to Case II of Section 159, but avoids its faults.

(c) In other "riding cut-off" gears, this peculiar arrangement of rockers of the Buckeye gear is not used, but both valves receive motion direct from their respective eccentrics. The general arrangement of the valves is shown in Fig. 174, in which both valves are external. The main valve has a false

end B, the only purpose of which is to provide an edge F, with respect to which the cut-off valve opens or closes. Each valve is driven by its own eccentric, the location of which is shown in

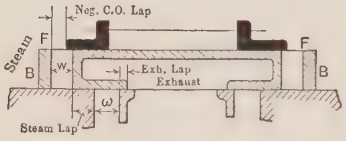


Fig. 174.

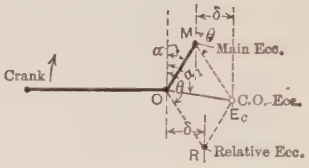


Fig. 175.

Fig. 175. The main valve controls the admission, release, and compression, and is designed as in (c) of Section 158. To analyze the action of the cut-off valve, its motion with respect to the main valve must be considered.

(d) As the crank revolves, not only do the eccentrics rotate about the center of the shaft, but they rotate about each other. That this last statement is true can be seen by turning Fig. 175 about O, and it will be noticed that, at the same time, the cut-off eccentric rotates about the main eccentric and in the same direction as that in which the crank turns. Evidently, then, *the motion of the cut-off valve with respect to the main valve is produced by the rotation of the cut-off eccentric about the main eccentric;* hence, *this motion is equivalent to that which a simple valve would have, with respect to a fixed seat, if driven by an eccentric having throw equal to the distance between the eccentric centers* (Fig. 175) *and with angle of advance equal to* θ. This imaginary eccentric will be called the "**relative eccentric**" R. Thus, to analyze the action of the cut-off valve with respect to the main valve (its moving seat), the position of the relative eccentric must first be determined, after which the valve diagrams would be constructed in the usual manner, but using the throw and angle of advance of this eccentric.

(e) The **Meyer Valve Gear** has the same arrangement of valves as that shown in Fig 174. The cut-off eccentric usually has a 90-degree angle of advance, that is, it is located opposite the crank. The cut-off is varied by changing the lap of the valve by the method shown in Fig. 167.

(f) The **Russell Valve Gear** is similar in character to Figs. 174 and 175, but the cut-off is altered by changing the position

of the cut-off eccentric. This adjustment is made by a shaft governor which turns the eccentric on the shaft to vary the angle of advance.

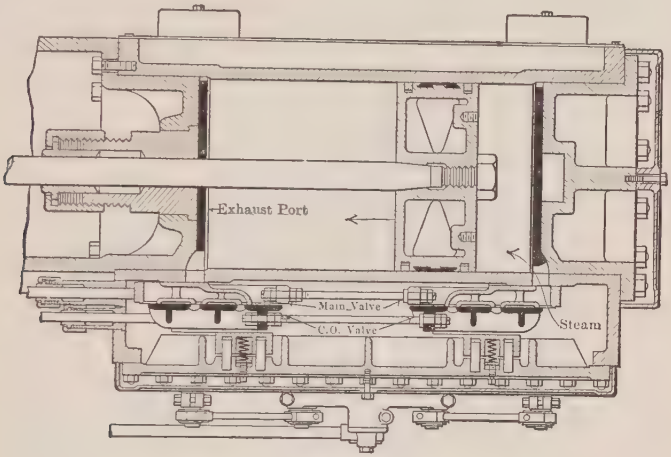


Fig. 176.

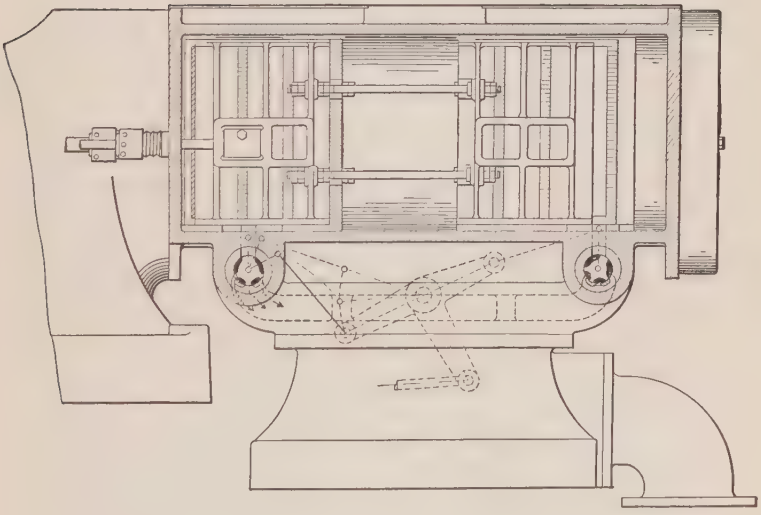


Fig. 177.

The valves are shown in Figs. 176 and 177. The main valve operates the admission only. The exhaust is controlled by separate triple-ported valves of the Corliss type shown in Fig. 177.

The cut-off valve is made triple-ported, as is its seat on the back of the main valve. The arrangement of eccentrics is similar to that shown in Fig. 170.

(g) The **McIntosh-Seymour Gear** has separate main, cut-off, and exhaust valves, of the "*gridiron*" type, working across the cylinder, as shown in the section in Fig. 178. These six

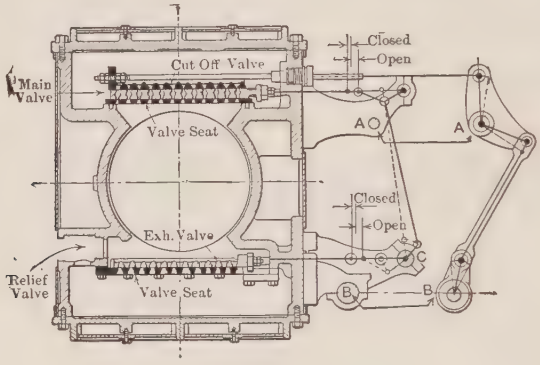


Fig. 178.

valves are driven by arrangements of rockers and toggles in the linkage, which distort the movements, so that after the valves are closed they have little motion; hence the friction and wear are reduced to a minimum.

The main valve receives its motion from a fixed eccentric, and the cut-off valve is driven by an eccentric which is turned about the shaft by a shaft governor to adjust the cut-off. Fig. 100 (p. 247) shows the general arrangement of the valve gear and the rocker shafts, which latter are given an oscillatory motion by the eccentrics acting through bell cranks. The arrangement of eccentrics is similar to Fig. 170.

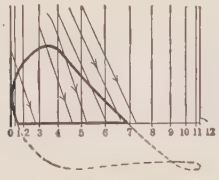


Fig. 179.

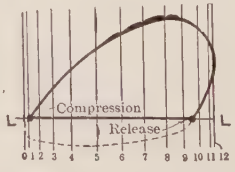


Fig. 180.

Fig. 179 is the distorted elliptical diagram for the main valve, with the opening diagram shown by bold lines. Superimposed on

the latter are lines showing the closure of the cut-off valve. Fig. 180 gives the distorted elliptical diagram for the exhaust valve. It is seen that the valve movements after closure are much less than with ordinary valve gears.

(h) There are many other possible arrangements of riding cut-off gears, a great number of which are in actual use.

161. Gears with Oscillating Valves. (a) Instead of having the slide valve flat, it may have a curved face, as in Fig. 181, in which case the valve oscillates about center O'. The displacement x of the eccentric-rod pin U from the Y-axis is always equal to that of the eccentric with respect to the vertical axis through the shaft. This valve is substantially equivalent to the ordinary flat D-valve, and would be designed or analyzed in

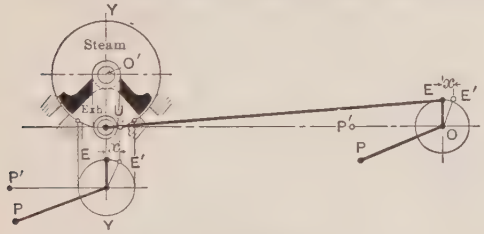


Fig. 181. — Oscillating Valve.

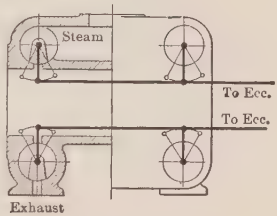


Fig. 182.

the same manner, using the same valve diagrams. These diagrams show the true positions of the crank for all events; but the laps, displacements, and openings are chordal, that is, would be measured as chords instead of as arcs.

(b) This arrangement of valve introduces very long steam passages, extending from the center to the ends of the cylinder, and this is not conducive to economical performance, as has already been seen.

(c) A better arrangement is one in which there are four oscillating valves, as in Fig. 182, each of which performs the single function of one of the four edges of the single valve. In the figure the outer edges of the two upper valves control the steam events, and the inner edges of the lower valves operate the exhaust events. The other edges of the valves perform no function. The chordal laps would be the same as in the case of the single valve of Fig. 181. The valves shown in Fig. 182 are of the "Corliss" type. With this arrangement the steam and exhaust passages are very short and direct, thus the clearance volume and surfaces are relatively small.

(d) All four valves may be driven by the single variable eccentric with shaft governor, as is common with high-speed engines. It is better, however, to connect the steam valves in this manner, and to drive the exhaust valves by a separate fixed eccentric, so that release and compression will remain constant.

(e) One fault of most valve gears is that the valve has large movement after it has closed. To reduce wear and friction, the movement should cease as soon as the overlapping is sufficient to prevent leakage. Also, it is desirable to have more rapid movement of the valve after it opens than is obtained with the

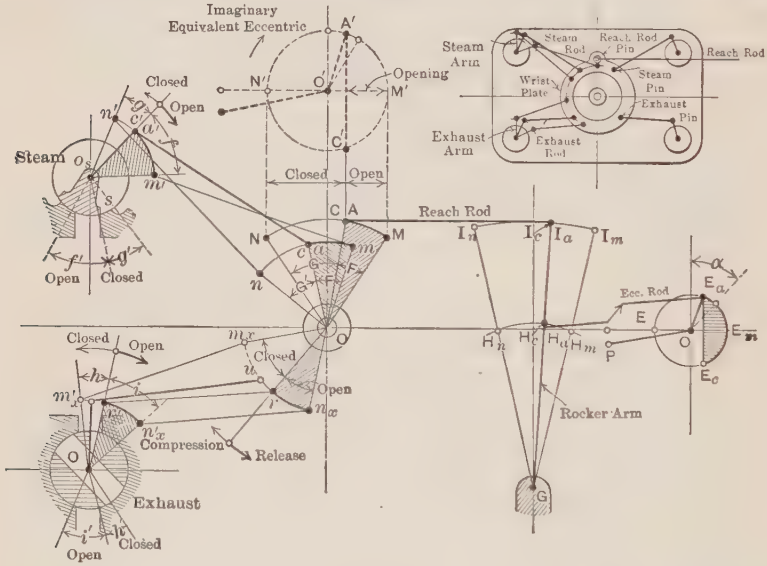


Fig. 183.

simple gear. Both results can be effected by using links and rockers so arranged as to give the valve the desired motion. One such arrangement is shown in the upper right-hand corner of Fig. 183. Engines using this type of gear may be called **High-Speed Corliss Valve Engines**, or positive cut-off Corliss engines.

(f) At the left of Fig. 183, the edge s of the steam valve is shown even with the port edge with which it opens or closes. Let g' be the desired (*small*) angular movement after closure, and f' be the (*large*) angle after opening. The steam arm $o_s a'$, which moves the valve, will swing through the same angles as the valve; thus g and f are respectively equal g' and f'. The

pin position a' for admission (motion to right) of course coincides with c' for cut-off (motion to left).

Starting at the right of the figure, E_a is the eccentric position for admission; E_m, that for maximum opening; E_c that for cut-off; and E_n is for extreme closure. The similarly subscripted positions of pins H and I on the rocker arm, the positions A, M, C, and N of the reach-rod pin, and a, m, c, and n of the steam pin, all respectively correspond with these eccentric positions. In each case the position for admission (motion to the right) coincides with the position for cut-off (motion to the left). It will be seen that the angular movement am of the steam pin for opening is smaller than that for closure cn, which is just contrary to what is desired for the valve movement. However, it is possible to locate the steam pin on the wrist plate in such position, and to use such a length of steam rod, that the steam arm moves through angles $n'c'$ and $a'm'$ respectively when the steam pin moves through angles nc and am, and thus to accomplish the desired result. With such arrangement the distances nn', aa', and mm' must of course all be equal, since they represent the length of the steam rod.

The exhaust valve motion can be similarly distorted so as to be small after closure and large after release. The arrangement for the crank end of the cylinder is identical except reversed.

(g) There are many other arrangements of linkage used for high-speed Corliss valve engines. Some involve the use of a separate fixed eccentric to drive the exhaust valves and thus obtain constant release and compression.

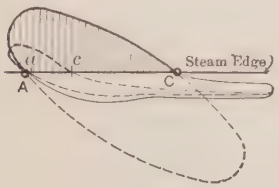


Fig. 184.

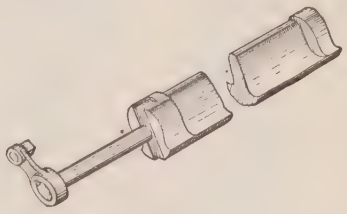


Fig. 186.

(h) In Fig. 184 is given the distorted elliptical diagram for the steam valve of one gear of this high-speed Corliss type.

(i) **The Trip-Cut-off Corliss Engine with Single Eccentric.** Fig. 185 is similar to the arrangement just discussed, except that the eccentric is fixed and the cut-off is operated by a tripping

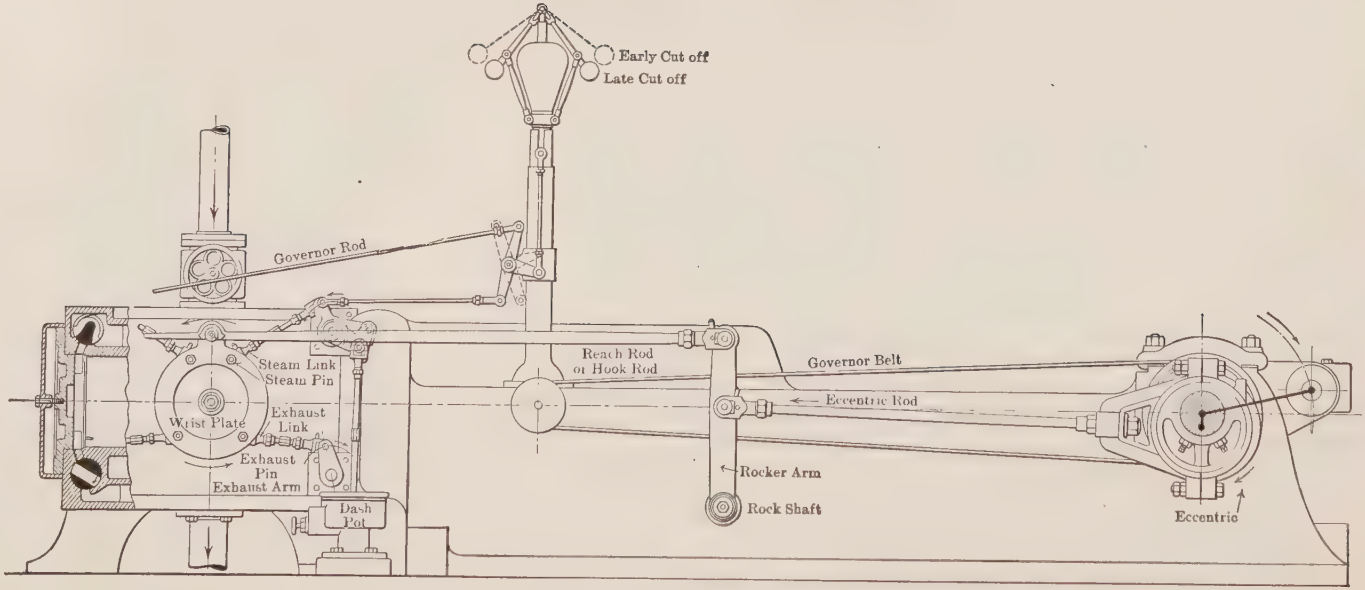


Fig. 185. — Corliss Valve Gear.

device positioned by a governor of the fly-ball type. The type of steam valve used is shown in Fig. 186; and the bonnet for the head-end steam valve, and the part of the gear which it supports, are illustrated in Fig. 187 (*a*). The names of these parts are given in Fig. 187 (*b*). The left arm *A* of the bell crank carries a hook *C*

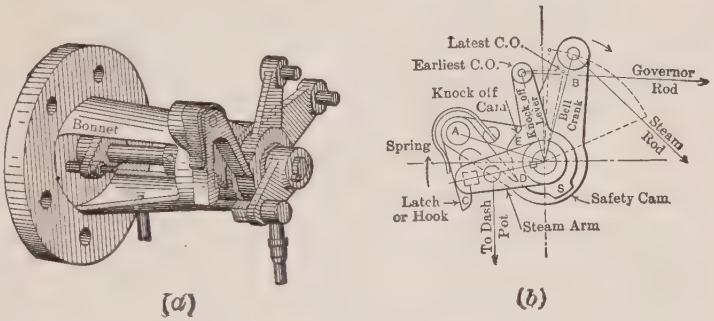


Fig. 187. — Steam Gear — Corliss Engine.

which engages with the steam arm on the valve stem. If the hook remains latched, the motion which the bell-crank arm *B* obtains from the wrist plate by means of the steam rod is transmitted directly to the valve, and the case is identical with that discussed in (f) of this section. In these engines, however, the governor controls the position of the knock-off cam *E*, which has a definite position corresponding to each different cut-off. As the bell crank is moved clockwise, the hook turns the steam arm and opens the valve (as in Fig. 183). This continues until the part *D* of the hook comes in contact with the *stationary* knock-off cam *E*, when the hook becomes disengaged from the steam arm, which is then returned to its lowest position by the dash pot, thus closing the valve.

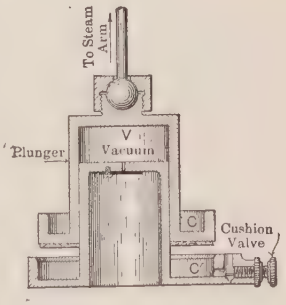


Fig. 188. — Dash Pot.

(j) The simple elements of the dash pot are shown in Fig. 188. When the steam valve is opened the plunger is raised and a vacuum is formed at *V*. After the hook has been tripped this vacuum causes the descent of the plunger and closure of the steam valve. The fall is stopped by the air cushion which is formed between *C* and *C'* and which is adjusted by the cushion valve.

(k) Fig. 189 shows the distorted elliptical diagram for the steam valve. With the trip occurring at t, the cut-off is at c. As the valve cannot close instantaneously, tc will slope somewhat. A similar diagram for the exhaust valve is given in Fig. 190.

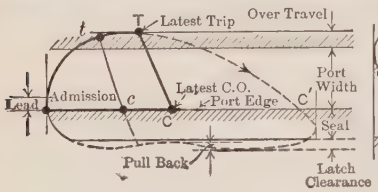


Fig. 189.

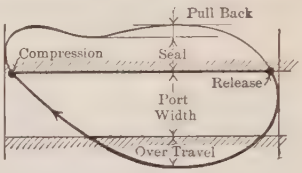


Fig. 190.

In connection with Fig. 187, it will be noticed that the trip of the head-end steam valve occurs when the hook end D comes in contact with the stationary cam E, while the hook and bell crank are *still moving to the right* (that is, *before the eccentric has reached* the *R.H. dead center*); and that if, when the eccentric arrives at this position, the trip has not taken place (E being too far to the right), it will not take place at all, and cut-off will occur at C' in Fig. 189 instead of at C.

(l) The angle of advance is fixed by the release and compression, as in the case of the main valve of the riding cut-off gears (Fig. 163). It is in no way dependent on the other events, for, with crank on dead center, the steam rod can be adjusted to give the valve the proper lead, and cut-off is controlled by the knock-off cam independently of the eccentric.

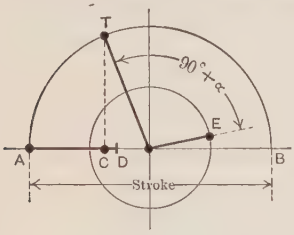


Fig. 191.

If E in Fig. 191 is the eccentric position for latest trip, the crank pin is then at T and the piston at C. As, however, some time must elapse before the valve is closed, cut-off will occur when the piston has reached some position D, which is usually at about 0.4 stroke. Thus with an ordinary *single eccentric Corliss gear the latest cut-off possible is about* 0.4 *stroke*, and this is accomplished only by using the smallest angle of advance that will give the proper release and compression.

(m) There are many other arrangements of valves, of trip gear, of wrist-plate linkage, and of dash pot, but all operate in

a manner similar to that described. The valves are frequently double- or multiple-ported.

(n) One of the faults of this gear is that a failure of governor belt stops the governor and lets the weights drop to the lowest position, thus advancing the cut-off to the latest point. The power then developed is greater than that absorbed, and the engine will "run away" and be demolished, unless stopped by hand or by some safety device.

One safety device consists of a "safety cam" *S* in Fig. 187 which prevents the hook engaging with the steam arm when the governor occupies its lowest position. Some engines have auxiliary fly-ball governors which will close the throttle valve when the speed becomes unsafe. There are many other forms of safety devices in use.

(o) The limitation of the latest cut-off can be avoided by using the **Two-Eccentric Corliss Gear,** in which one eccentric drives a wrist plate for the exhaust valves, and another one actuates the steam wrist plate. Fig. 192 shows the arrangement of eccentrics, with crank on dead center. The angle β between the crank and steam eccentric fixes the latest cut-off, but with this arrangement, since the exhaust valves are driven independently, it may be made any value within limits. The angle used in the figure permits of cut-off as late as three-fourths stroke as is seen from the extreme (dotted) position.

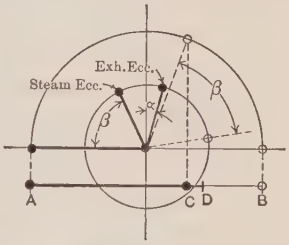


Fig. 192.

Late cut-off can also be obtained by using a moving knock-off cam which may be oscillated either by a separate small eccentric at about 90 degrees with the main eccentric, or by the sidewise motion of the eccentric rod, which is 90 degrees out of phase with the longitudinal movement. With such arrangement the knock-off cam overtakes the hook and releases it even after the main eccentric has rotated a considerable angle beyond the dead-center position.

(p) The rotative speed of trip-cut-off Corliss gears must be relatively low, for otherwise the hook gear becomes uncertain in action. Speeds above 120 r.p.m. are seldom used, and generally they are considerably less. Hence engines using this type of gear are commonly classified as "low-speed."

(q) There are several trip-cut-off gears which have gridiron valves working across the cylinder either horizontally (somewhat similar in arrangement to Fig. 178) or vertically. Trip-cut-off gears are also used with poppet valves (Section 164).

162. Link Gears. (a) The valve gear most commonly used on engines which are reversed is the **Stephenson Link Gear**, one arrangement of which is shown semi-diagrammatically in Fig. 193. The illustration is for a vertical engine with cylinder above, but the arrangement for a horizontal engine would be identical except for the position of the longitudinal axis.

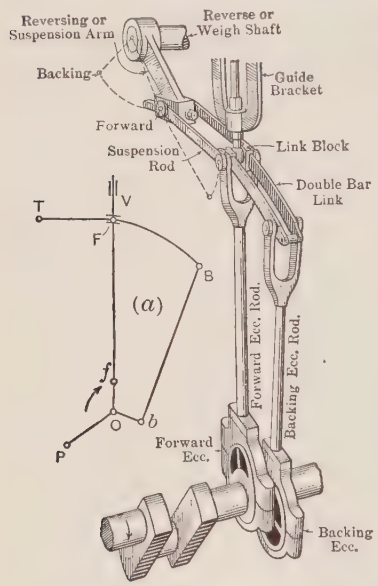


Fig. 193. — Stephenson Link Gear.

(b) The eccentrics are arranged as in Fig. 194, with the "forward eccentric," *f*, placed 90 degrees plus angle of advance

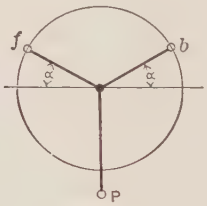


Fig. 194

ahead of the crank in the forward direction of rotation, and the "backing eccentric," *b*, at the same angle in the opposite direction. If the valve receives all its motion from eccentric *f*, the rotation will be forward (clockwise in this case); if from *b*, it will be backward (counter-clockwise).

In Fig. 193 it is seen that the forks at the ends of the two eccentric rods are connected by a "link" (whence the name of this type of gear), different points of which may be brought opposite the "link block" on the end of the valve stem, by turning the "reverse" shaft. The illustration shows the forward end opposite (in "full gear" forward), hence the valve is receiving all its motion from the forward eccentric and conse-

quently rotates forward with latest cut-off possible. If the other end of the link is brought opposite the link block ("full gear" backing), the engine would operate backward at maximum cut-off.

With the middle of the link opposite (" mid-gear "), the valve receives motion equally from both eccentrics; and the valve will open an amount equal to the lead and close immediately, the cut-off being practically at zero stroke.

If the link is shifted from mid-gear toward the forward end, the valve will still receive motion from both eccentrics, but the major part will be from the forward eccentric. As the forward end of the link is shifted nearer the link block, the width of valve opening is increased and the cut-off is advanced in a manner quite similar to that in the gear with single variable eccentric, Figs. 160 and 161, when the eccentric is moved from inner position *3* outward towards *1*. In fact, an approximate analysis of the Stephenson link gear can be made by considering the valve as driven by a single swinging eccentric with a radius of path R which can be computed by McFarlane-Gray's formula:

$$R = \frac{\text{distance between eccentric centers} \times \text{length of ecc. rod}}{2 \times \text{distance between eccentric-rod pins}}. \quad (275)$$

(c) If, with the crank P pointing away from the cylinder, the rods are not crossed, as in Figs. 193 and 195 (*a*), the arrangement is termed **" open rod."** In this case the path of the equivalent single eccentric is *feb* with radius R. If, with crank in the same position, the rods are crossed, as in Fig. 195 (*b*), it is a **" cross-rod " linkage,** and the path of the equivalent eccentric is *feb*.*

For any link position, the equivalent eccentric occupies the same relative position on its path *fb* as the link block on the link *FB*. It is seen that the open-rod linkage gives increasing lead as the cut-off is decreased, whereas the reverse occurs with crossed rods. From Eq. (275) it is seen that using longer eccentric rods increases R, thus making the path straighter and the lead less variable. To have the lead vary equally at the two ends of the valve, the radius of the link arc must equal the length from eccentric center to eccentric-rod pin, in the arrangement shown.

* Note that when the crank has rotated 180 degrees the rods are crossed in the "open-rod" arrangement and open in the " crossed-rod " gear. In classifying the arrangement the crank *must point away from the cylinder.*

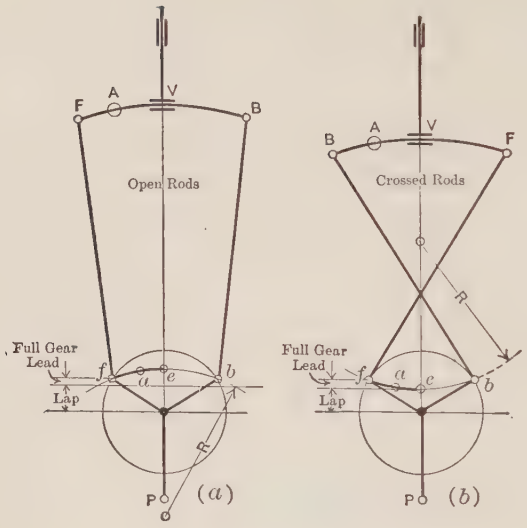


Fig. 195.

(d) The link shown in Fig. 193 is of the "double-bar" type. There are many other arrangements: some have the eccentric-rod pins offset from the link; on some the suspension-rod pin is located at the middle of the link arc; on others, between the middle and the end. The modifications introduced in such cases cannot be considered here.

(e) For the method of making an exact analysis of the action of the valve operated by a Stephenson link see textbooks on valve gears.

(f) In the **Gooch Link Gear**, Fig. 196, the "radius rod," instead of the link, is shifted to change the cut-off. As the link radius equals the length of the radius rod, there is no movement of the valve if the adjustment is made when the crank is on dead center, as in the figure; hence the lead is constant. Line $b'f'$ is the path of the equivalent single variable eccentric, and $b'bO$ is a right angle.

(g) The **Allan Link Gear** shown in Fig. 197 has a straight link. The link and the radius rod are shifted in opposite directions in such manner that the valve is not moved when crank is on dead center, hence the lead is constant. The path of the

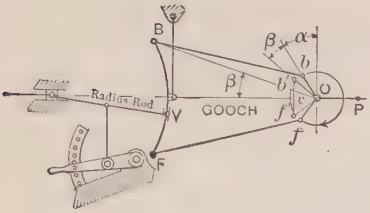


Fig. 196. — Gooch Link Gear.

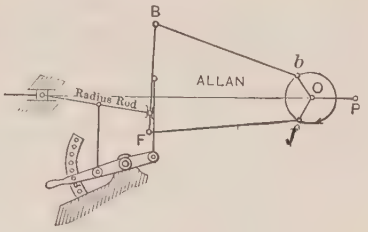


Fig. 197. — Allan Link Gear.

equivalent single eccentric is similar to that in the Gooch linkage.

(h) The **Porter-Allen Gear** shown in Fig. 198 has a link which is consolidated with the eccentric strap and is guided at *A* along

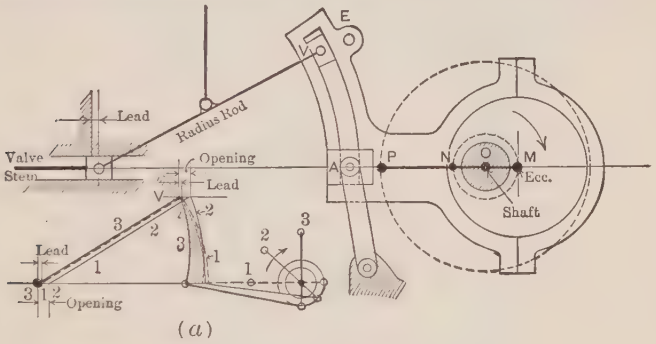


Fig. 198. — Porter-Allen Gear.

the center line of the engine. The throw *OM* of the eccentric equals lap plus lead, thus, in the position shown, the head end of the valve is open to lead. As the eccentric rotates from this position the tilting of the link increases the opening, which later is decreased by the translatory motion of the link. At (*a*) in the figure, with link block at *V*, maximum opening occurs at crank position *2* and cut-off at *3*. With link block in a lower position, there would be less opening and earlier closure, the lead remaining the same, however.

Motion satisfactory for exhaust valves can be obtained from some point such as *E*.

163. Radial Valve Gears. (a) In vertical multicylinder marine engines using link gears, the valves are usually located per-

pendicularly over the shaft, and some or all of them lie between the cylinders and thus lengthen the engine. It is true that by using rocker arms the valves might be placed at the side, but that arrangement of mechanism has certain disadvantages, and even then the eccentrics prevent, to a certain extent, the shortening of the engine.

Using the type of valve gears known as "radial gears" necessitates placing the valves on the side of the engine. In most of the gears of this type a single eccentric is used and in some the eccentrics are dispensed with altogether. With this type of gear the engine can be made to occupy less space than with link gears.

There are a great many kinds of radial gears; only the most important will be described.

(b) The **Marshall Type of Gear,** which is shown in Fig. 199, uses a single eccentric, either at 0° or 180° with the crank.

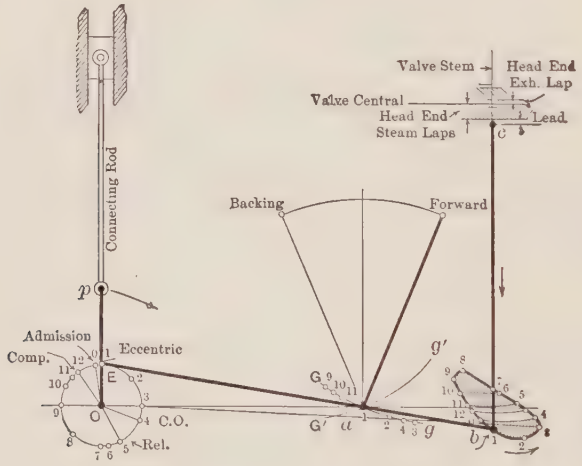


Fig. 199. — Marshall Type of Radial Gear.

The point *a* on the eccentric rod *Eab* is guided along path *Gg*.* The end *b* traces the oval figure shown, and its positions are numbered to correspond with those of the eccentric and crank. The motion which the valve receives through the rod *bc* is practically the same as that obtained from an eccentric. By changing the inclination of the guide *Gg* the oval is changed, the

* The Hackworth gear has a straight guide.

amount of opening is altered, and the cut-off is varied. A reversal of the inclination, as $G'g'$ reverses the engine. The pin b may either be located as shown or it may be between E and a.

(c) If any point in a linkage moves in phase with the crank and describes a path that is approximately circular, a pin located at that point can be used instead of the eccentric to give the valve the motion equivalent to that obtained with the Marshall gear.

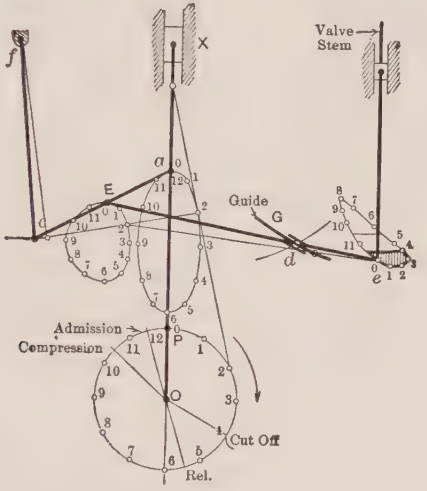


Fig. 200. — Joy Radial Gear

In the **Joy Gear,** shown in Fig. 200, ac is a link with one end attached to the connecting rod and the other end to the suspension link fc. The point E moves in a path which may be substituted for the eccentric circle. The rest of the linkage resembles the Marshall in character and performance.

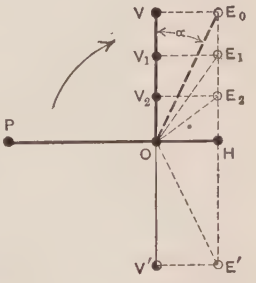


Fig. 201.

(d) If, in Fig. 201, the harmonic motion received from an eccentric H opposite the crank is combined with that from another eccentric V at right angles to the first, the resultant motion is equivalent to that which would be obtained from an eccentric located at E_0 (found by constructing the parallelogram OVE_0H),

and a valve receiving this combined motion would operate satisfactorily. OH is made equal to the lap plus lead, and OV may be varied to change the angle of advance and throw of the equivalent eccentric E_0, which has HE_0 as its path and resembles the single variable eccentric previously discussed.

The Walschaert Valve Gear shown in Fig. 202 uses this principle. If the link block d is shifted to the middle e of the link, point c will remain practically stationary. Then the lever ab,

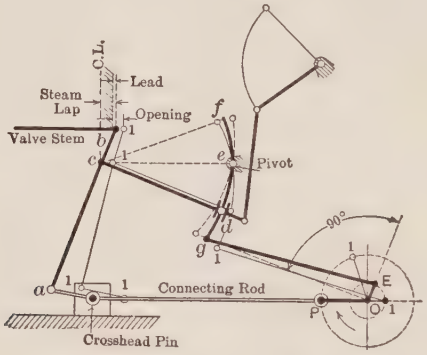


Fig. 202. — Walschaert Radial Gear.

which is pivoted at c and receives motion at a from the crosshead, will vibrate in such manner that the end b will displace the valve a distance equal to lap plus lead when the crosshead reaches the end of its stroke, and that the valve motion will equal that received from eccentric OH in Fig. 201. The link fg receives motion from an eccentric E, which is 90 degrees out of phase with the other motion. With link block in any position d (other than central) on the link, point c, and consequently b, will receive this motion, which is equivalent to that obtained from eccentric OV in Fig. 201. The resultant motion of the valve is that which would be given by the eccentric OE_0.

By shifting the link block d the amplitude of its motion can be varied, and this is accompanied with corresponding change in the width of valve opening and time of cut-off. If shifted above the pivot e, the engine would be reversed.

The Walschaert gear is widely used on locomotives of the largest sizes. Being located on the outer side of the engine, it places no limitation on the size of the boiler, as does the Stephen-

son link gear, which is located directly below the boiler and requires considerable room for shifting from one full-gear position to the other.

164. Poppet Valves and Their Gears. (a) Poppet-lift valves (Figs. 203 and 204) have no friction nor wear from sliding.

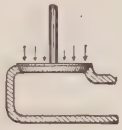


Fig. 203

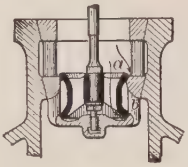


Fig. 204.

They require no lubrication, and being symmetrical do not warp with temperature changes; hence they are suitable for use with

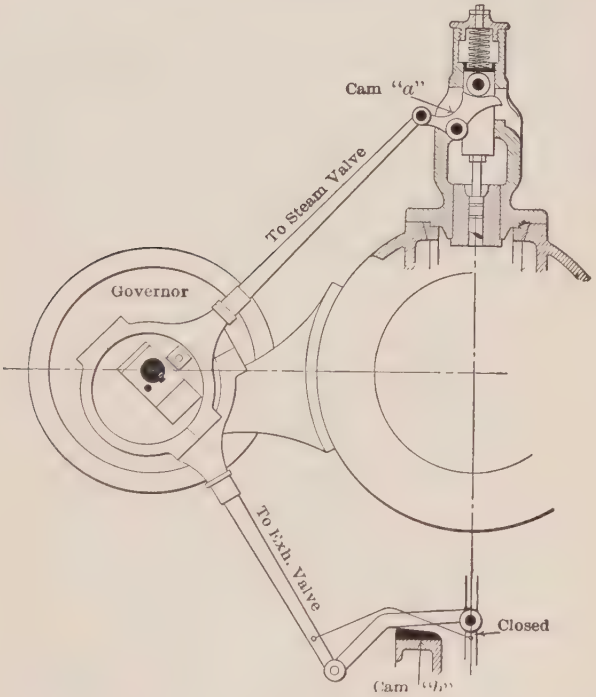


Fig. 205.

highly superheated steam. The ordinary single poppet or mushroom valve, Fig. 203, is hard to open because of the un-

balanced pressure on its back; therefore, the double-seated type of valve, one form of which is shown in Fig. 204, is commonly used instead, since the steam pressures on upper and lower sides are about equal.*

There is a great variety of arrangements of such valves and of their gears.

(b) The valve may be operated by a continuously **rotating cam;** and there may be a sleeve with variable cam surface which may be moved endways to change the valve events.

(c) An **oscillating cam,** as *a* in Fig. 205, may be used, and it may be driven by an eccentric which is shifted by a shaft type of governor, as in the figure; or it may be driven by a fixed eccentric, in which case the cut-off may be operated by trip or by shifting the cam, or by changing some intermediate linkage to distort the motion; or closure may be brought about by some other means.

(d) The valve may be operated by a **floating lever** which fulcrums on a cam surface, as *b* in Fig. 205, and which is driven by an eccentric, which may be variable or stationary. The cut-off can be changed by any of the methods given in (c).

(e) Cams are also used to operate other types of valves, such as piston valves and gridiron valves.

* Allowance must be made for the area of the valve stem.

CHAPTER XX.

CONVENTIONAL INDICATOR DIAGRAM.

165. Conventional Diagram for Simple Engines. (a) If the actual indicator diagram has been obtained from an engine, the m.e.p. may be determined by any of the methods discussed in Section 102, and the i.h.p. of the engine may be obtained by using Eq. 210. In making such computation for a double-acting engine, however, the *area of the piston rod* must be deducted from the area of the piston on one side, and the average of the areas on the two sides of the piston must be used in the formula; or else the i.h.p. for each side of the piston must be computed separately.

(b) When actual indicator diagrams are not available, it is customary to use a conventional diagram, with proper correction factor, for estimating the probable m.e.p.

(c) Before the conventional diagrams can be drawn, however, the *clearance volume* in the cylinder must be known. This volume can be determined by pouring a measured quantity of water into the clearance space. It can also be found *approximately* from the actual indicator diagram in the following manner (shown in Fig. 206): Select two points 1 and 2 on the expansion line and draw a rectangle with these points as corners and with the sides parallel to the respective PV-axes. Then, the diagonal through the other corners will cut the V-axis at the origin *O*, assuming that the expansion equation is PV = constant. Then *Cl* in the figure is the clearance volume to scale. The compression curve may be used in a similar manner to find *O*. This makes application of the construction shown in Fig. 11.

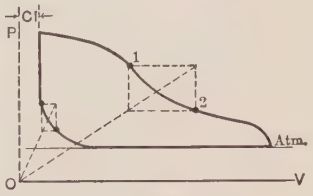


Fig. 206.

The clearance volumes used in practice are about as follows:

Single-valve engine	5 to 15%
Multi-valve engines	2 to 8%

(d) In constructing conventional diagrams for estimating the probable power of an engine, it is customary to assume that expansion follows the equation $PV = P_1V_1 = P_2V_2 = \text{constant}$, instead of being adiabatic. This is because the "equilateral hyperbola" is easier to construct than the adiabatic curve, and because the actual expansion line follows it as closely as it does the latter. The expansion line may be constructed by the methods shown in Figs. 11 and 12.

The foot-pounds of work (A) represented by the area (Fig. 207) under such an expansion line is found in the manner already discussed in Section 29 (c) to be

$$A = \int_{V_1}^{V_2} P\,dV = P_1V_1 \int_{V_1}^{V_2} \frac{dV}{V} \quad . \quad . \quad . \quad . \quad (276)$$

$$= P_1V_1 \log_e \frac{V_2}{V_1} = P_1V_1 \log_e r,^* \quad . \quad . \quad . \quad . \quad (277)$$

where r is the ratio of expansion $\left(\frac{V_2}{V_1}\right)$. In Fig. 208 r is $\frac{V_c}{V_b}$.

(e) In the case of an **engine without clearance** the conventional diagram is *abcde* of Fig. 208. The work shown by area

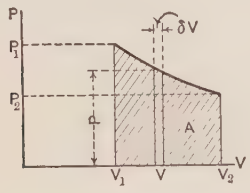


Fig. 207.

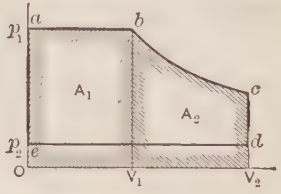


Fig. 208.

A_1 is P_1V_1 foot-pounds, and that represented by A_2 is $P_1V_1 \log_e r$. Hence if the back pressure is P_2, the work shown by the conventional diagram *abcde* is

$$P_1V_1 + P_1V_1 \log_e r - P_2V_2 = P_mV_2,$$

in which P_m is the mean effective pressure. Solving this equation for P_m gives

$$P_m = P_1\left[\frac{1 + \log_e r}{r}\right] - P_2.$$

As the m.e.p. is generally used in pounds per square inch, it is

* $\text{Log}_e = 2.302 \log_{10}$.

more convenient to divide both sides of this equation by 144, giving

$$p_m = p_1\left[\frac{1+\log_e r}{r}\right] - p_2. \quad . \quad . \quad . \quad . \quad . \quad (278)$$

The values of the bracketed quantity for different values of r are given in Table V.

TABLE V.

r	$\frac{1+\log_e r}{r}$	r	$\frac{1+\log_e r}{r}$	r	$\frac{1+\log_e r}{r}$
1.0	1.00	6.0	0.465	16.0	0.236
1.5	0.937	7.0	0.421	17.0	0.226
2.0	0.847	8.0	0.385	18.0	0.216
2.5	0.766	9.0	0.355	19.0	0.208
3.0	0.700	10.0	0.330	20.0	0.200
3.5	0.644	11.0	0.309	21.0	0.192
4.0	0.597	12.0	0.290	22.0	0.186
4.5	0.556	13.0	0.274	23.0	0.180
5.0	0.522	14.0	0.260	24.0	0.174
5.5	0.492	15.0	0.247	25.0	0.169

(f) The actual indicator diagram of course differs from the computed one drawn by this method. The ratio of the area of the actual to that of the conventional diagram is called the **" Diagram Factor "** (DF). Then if the diagram factor is known for engines similar to that which is being considered, the probable m.e.p. for the new engine is

$$p_m' = DF \times p_m. \quad . \quad . \quad . \quad . \quad . \quad . \quad (279)$$

It is a common practice to use Eq. (278) even for engines which have clearance, and to substitute $\left(\frac{1}{cut\text{-}off\ ratio}\right)$ for r, thus ignoring the clearance.

The diagram factors to be used for different types of engines in such cases are given in the following table:

TABLE VI. — DIAGRAM FACTORS.

Simple slide-valve engine	55 to 90%
Simple Corliss engine	85 to 90
Compound slide-valve engine	55 to 80
Compound Corliss engine	75 to 85
Triple-expansion engines	55 to 70

(g) The conventional diagram for an **engine with clearance** is shown by *abcde* in Fig. 209. Here the ratio of expansion is

$$r = (L + l_c) \div (l + l_c), \quad . \quad . \quad . \quad . \quad . \quad (280)$$

using scalar distances to represent volumes.

The net work shown by the area is

$$A = A_1 + A_2 - A_3$$
$$= P_1 l + P_1 (l_c + l) \log_e r - P_2 L. \quad . \quad . \quad . \quad (281)$$

Dividing by L and by 144 gives the mean effective pressure for this case as follows:

$$p_m = \frac{A}{144\,L} = p_1 \left\{ \frac{l + (l_c + l) \log_e r}{L} \right\} - p_2. \quad . \quad . \quad (282)$$

To simplify this expression, let $C = \dfrac{l}{L}$ = cut-off ratio, and $\dfrac{l_c}{L} = c$ = clearance ratio; then

$$p_m = p_1 \{C + (c + C) \log_e r\} - p_2. \quad . \quad . \quad . \quad (283)$$

The diagram factors for this case are 3 or 4 per cent larger than those given in Table VI.

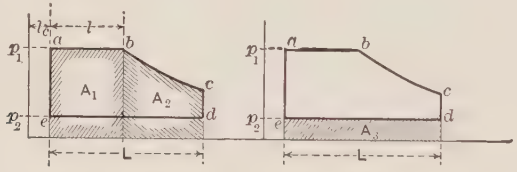


Fig. 209.

(h) **With compression,** the diagram of Fig. 209 is reduced by the area D in Fig. 210.

If p_f is the pressure at the end of compression, the reduction of the m.e.p. caused by this small area is evidently

$$p_{mD} = p_f \left(\frac{l_c}{L}\right) \log_e \left(\frac{l_k + l_c}{l_c}\right) - \frac{p_2 l_k}{L};$$

from which, since $\quad p_f = \dfrac{p_2 (l_k + l_c)}{l_c},$

$$p_{mD} = p_2 \left(\frac{l_k + l_c}{L}\right) \log_e \frac{l_k + l_c}{l_c} - \frac{p_2 l_k}{L}.$$

Subtracting this from Eq. 282 and letting k represent the compression ratio $\left(\dfrac{l_k}{L}\right)$, gives the m.e.p. of diagram *abcdef* as

$$p_m = p_1 \{C + (C + c) \log_e r\} - p_2 \left\{ 1 - k + (k + c) \log_e \frac{k + c}{c} \right\}. \quad (284)$$

In this case the diagram factors are 4 to 6 per cent larger than the values given in Table VI.

(i) A conventional diagram that approaches closer to the actual diagram than any that have been discussed is shown in Fig. 211. This has the sloping admission line. The area is made up of the triangle A and the area B, similar to that for

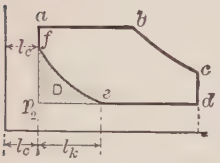


Fig. 210.

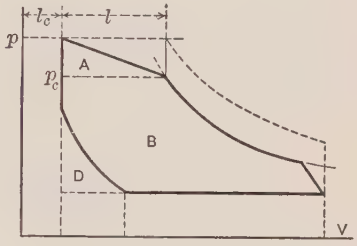


Fig. 211.

which Eq. 284 was developed, and much less correction is necessary for obtaining the probable m.e.p. than in the previous cases.

(j) For noncondensing simple slide-valve engines operating under ordinary conditions, with steam pressure about 100 pounds gauge, the m.e.p. at the most economical cut-off is about one-half the initial gauge pressure. For simple Corliss engines the m.e.p. is about four-tenths the initial gauge pressure, under the same conditions.

These values may be used only when the estimates are very approximate.

166. Conventional Diagrams for Multiple-Expansion Engines. (a) By referring to Fig. 93, on which diagrams of both the high-pressure and the low-pressure cylinders of a compound engine are drawn to the same scale, it will be seen that if the dividing line at T_R is omitted, there results a single indicator diagram of area equal to the sum of the areas H.P. and L.P.; thus, theoretically, a simple engine of the same size as the lower-pressure cylinder (total volume = V_2) would give the same amount of power with this single diagram as is obtained with the two cylinders of the compound engine.

Evidently, then, to calculate the i.h.p. of the compound engine, it is only necessary to consider the m.e.p. of this simple

(or "combined") diagram as acting on the low-pressure piston. The i.h.p. of triple- and quadruple-expansion engines can be computed in a similar manner.

The m.e.p. of the combined diagram is usually called the "m.e.p. referred to the low-pressure cylinder," or more briefly the **"referred m.e.p."** Its theoretical value can be computed by Eqs. (278), (283), or (284), and the probable m.e.p. is found by correcting with the diagram factor. Values of the latter are given in Table VI for use with Eq. (278). Modified values should be used with Eqs. (283) and (284).

(b) If it is desired to estimate the size of a compound engine that will give a specified amount of power, the referred m.e.p. is first computed; then with the stroke, L (feet), and number n of cycles per minute selected, the *area of the low-pressure piston* (square inches) to give any i.h.p. can be computed from

$$a_L = \frac{\text{i.h.p.} \times 33{,}000}{p_{mR} \times DF \times L \times n}, \quad . \quad . \quad . \quad . \quad . \quad (285)$$

in which DF is the diagram factor (see Table VI).

Then with the ratio R of low-pressure cylinder volume to that of the high-pressure cylinder known, the *area of the high-pressure piston* is of course $1/R$th of the low-pressure area if the strokes are equal.

The size of the cylinders in triple- and quadruple-expansion engines is found in similar manner. The cylinder ratios to be used are found in Section 170.

(c) The diagrams of multiple-expansion engines will now be considered more in detail, and to facilitate the discussion the engines will be divided into two groups: (1) the Woolf type, without receivers; and (2) engines with receivers.

167. Diagrams of Woolf Type of Engine. (a) The compound engine was patented in 1781 by Jonathan Hornblower, but Watt's broad patents on expansion steam engines delayed its use. In 1804, Woolf reintroduced the compound engine and used an arrangement in which the steam was exhausted from the high-pressure cylinder, directly through very short passages to the low-pressure cylinder. Because there is little or *no receiver* volume or storage volume between the cylinders in such an engine, it is necessary for the pistons to start and finish their strokes together, and the *low-pressure cylinder must receive steam*

throughout its entire stroke from the high-pressure cylinder. If the steam were cut off in the low-pressure cylinder, there would be no place into which the high-pressure steam could be exhausted during the remainder of the stroke after this cut-off had occurred. The pistons may move together or in opposite directions.

(b) Fig. 212 (*a*) shows a Woolf engine whose pistons move synchronously and in the same direction. This motion would result if both piston rods were connected to the same end of a "walking beam" or to cranks set together. In Fig. 212 (*b*) the indicator diagram *H* is for the head end of the high-pressure cylinder and *L* is for the crank end of the low-pressure cylinder, clearance volume being neglected in both cases. In operation, steam is admitted to the high-pressure cylinder according to line *abc*; it is cut off at *c*; is expanded along *cd*; and it is exhausted from the high-pressure cylinder along line *da*. This steam exhausted from the high-pressure cylinder is received by the low-pressure cylinder along the line *ABC* and is then exhausted along line *CDA*. *BC* and *da* will be called hereafter the **line of transference** or **receiver line.** In Fig. 212 (*b*) the indicator cards of both cylinders have the same length, that is, the abscissas are piston positions, and are numbered to correspond with the positions shown in Fig. 212 (*a*).

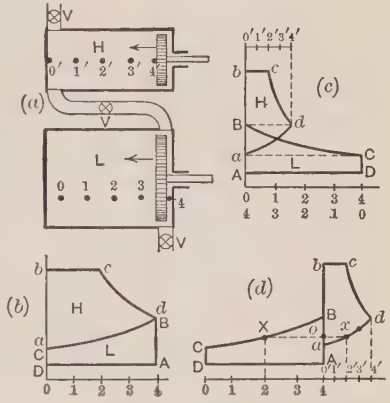


Fig. 212.

In Fig. 212 (*c*) the diagrams have been "combined," with abscissas representing the respective *volumes* in the two cylinders. In Fig. 212 (*d*) the diagrams have been combined in such a way that the volume of the steam during transference from the first to the second cylinder can be scaled directly. Thus, when the pistons have reached simultaneous positions 2 and 2′ the distance 0′–2′ (= *ox*) is the volume of steam in the high-pressure cylinder, the distance 4–2 (= *oX*) is the volume it occupies in the low-pressure cylinder, and distance 2′–2 (= *Xx*) is the total volume of the steam between the two pistons for

this position in the stroke. Obviously, the distances between piston positions bearing like numbers in this figure represent the volumes of steam during the period of transference. After these volumes have been determined (by scaling), the pressures at the corresponding piston positions can be found if the expansion is assumed to be hyperbolic, for during expansions *cd*, *da*, and *BC* the product *PV* remains constant, since there is no change in the quantity of steam involved during these processes. Thus the high- and low-pressure PV-diagrams can be readily constructed.

168. Diagrams for Engines with Infinite Receivers and No Clearance (General). (a) If a receiver of infinite volume is placed between the cylinders of the Woolf engine the curves *da* and *BC*, in Fig. 212, would become horizontal straight lines, and the low-pressure indicator diagram would be a rectangle. Evidently, with finite receiver, the larger the receiver volume the more nearly horizontal and straight will the line of transference become.

With a receiver of considerable volume into which the high-pressure cylinder can exhaust, it is possible to "cut off" in the low-pressure cylinder and thus to expand the steam independently in this cylinder. The pressure of the receiver will vary, because part of the time steam is being received from the high-pressure cylinder, at other times steam is being delivered to the low-pressure cylinder, and during part of the cycle both of these operations may occur simultaneously. Consequently the back-pressure line on the H.P. indicator diagram and the admission line of the L.P. diagram will be irregular. The character of the line of transference will be discussed in detail later.

(b) When a receiver of considerable volume is used it is possible to have any angle between the cranks of the two cylinders, whereas in the Woolf engine this angle is limited to zero degrees or 180 degrees in cases where there is a separate crank for each cylinder.

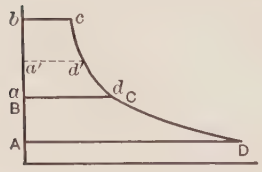


Fig. 213.

(c) In Fig. 213, *AbcD* is a conventional "combined" diagram for a compound engine with receiver of infinite volume. In it, *AD* is the volume

of the low-pressure cylinder, *ad* is that of the high-pressure cylinder, *bc* is the volume of steam admitted to the high-pressure cylinder, and *BC* is that at the time of cut-off in the low-pressure cylinder. Then

$\frac{ad}{bc} = r_H$ = ratio of expansion in the high-pressure cylinder;

$\frac{AD}{BC} = r_L$ = ratio of expansion in the low-pressure cylinder;

$\frac{AD}{bc} = r_T$ = total ratio of expansion.

Since, if hyperbolic expansion is assumed,

$$p_c\,(bc) = p_d\,(ad),$$

the receiver pressure is evidently

$$p_d = p_c\left(\frac{bc}{ad}\right) = p_c\frac{1}{r_H}, \quad . \quad . \quad . \quad . \quad . \quad (286)$$

for the case in which the expansion is complete in both cylinders.

(d) It is evident that the horizontal transfer line obtained with a receiver of infinite volume would correspond to the *mean* transfer pressure if a receiver of finite volume is used, and that indicator diagrams drawn with this horizontal transfer line would have practically the same areas as with the variable line of the small receiver. Hence these diagrams may not only be used for the engine as a whole but also when each cylinder is considered separately.

(e) **Changing the low-pressure cut-off** to make it occur earlier results (1) in raising the receiver line, as shown dotted in Fig. 213; it also results (2) in a *reduction* of the area of the high-pressure diagram and (3) an *increase* in the area of the low-pressure diagram. Making the low-pressure cut-off later reverses these results. Thus the cut-off in the low-pressure cylinder influences the receiver pressure and distribution of work between the cylinders, but does not affect the total work done by the engine.

(f) The **selection of the receiver pressure** is based on the following considerations:

(1) For greatest economy in the use of steam the *temperature ranges* in the two cylinders should probably be equal, although this is not certain. Hence the receiver pressure should probably be such that the corresponding temperature of the steam is

midway between the initial and final temperatures of the working fluid. Other considerations may be more important than this, however.

(2) It is sometimes desirable to have the *same cut-off* (that is, the same expansion ratios) in both cylinders. For example, in the tandem compound engine shown in Fig. 107, the two valves are on the same rod, hence the cut-offs in the two cylinders must change together.

(3) Usually it is desirable to have *equal work* done in the two cylinders. In this case the receiver line should be so drawn that the areas of the high-pressure and low-pressure diagrams are equal. This is especially desirable when the engine is a cross compound.

(4) In some special cases, equal *maximum thrusts* on the piston rods are desirable, and these thrusts are dependent on the receiver pressure.

(5) The *uniformity of turning effort* is dependent on the shape and relative proportions of the indicator diagrams of the two cylinders, and hence is dependent on the receiver pressure.

Usually compound engines are operated to perform equal work in the two cylinders, and this gives about as uniform a crank effort as is possible, and hence, considerations (3) and (5) are satisfied together with sufficient accuracy for ordinary purposes.

169. Receiver Pressures in Compound Engines with Infinite Receivers and No Clearance. (a) It has just been seen that the distribution of work among the cylinders depends on the receiver pressures, hence the problem is one of determining the mean receiver pressures which will give the desired distribution. The receiver pressure may be determined either graphically or analytically, using the conventional diagram. The *receiver volume* will be considered *infinite* and the *clearance zero*.

(b) The **graphical method** will be considered first.

Let p_1, p_2, and V_2 in Fig. 214 be given, and assume a terminal pressure p_D such as will give the drop (DE) in pressure in accordance with Section 111. With this data available, the combined PV-diagram, $AbcDE$, can be easily drawn and its work area can be determined. If the high-pressure cylinder is to do $1/n$th of the total work, the area H will be $1/n$th of the total area. The problem then is to find the location of line *ad* which

will give this distribution of work. The line *ad* can be drawn tentatively and then the area above it can be integrated by planimeter to see if it has the proper value. If it is not correct, another position of *ad* can be tested, and by repeated trials a proper receiver line can be obtained by this "cut and try" method. This same method can be used when the H.P. expansion is incomplete (i.e., when the toe of the H.P. diagram is removed) as in Fig. 215, and can also be applied to multiple-expansion engines with any number of expansion cylinders.

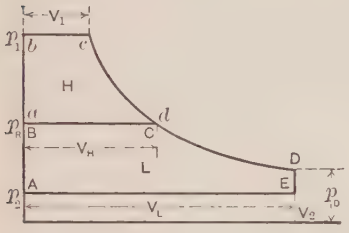


Fig. 214.

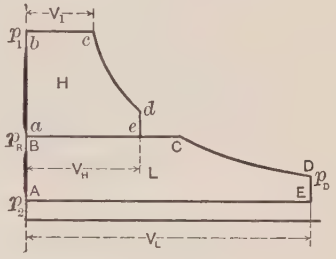


Fig. 215.

In Figs. 214 and 215, V_H is the volume of the high-pressure cylinder; and the corresponding mean effective pressure acting in the high-pressure cylinder is

$$p_{mH} = \frac{\text{area } H}{\text{length } V_H} \times \text{scale of ordinates.} \quad . \quad . \quad . \quad (287)$$

Similarly the L.P. mean effective pressure is

$$p_{mL} = \frac{\text{area } L}{\text{length } V_L} \times \text{scale of ordinates.} \quad . \quad . \quad . \quad (288)$$

The total m.e.p. "referred" to the low-pressure cylinder is

$$p_{mR} = \frac{\text{area } (H + L)}{\text{length } V_L} \times \text{scale of ordinates.} \quad . \quad . \quad (289)$$

(c) By removing the toe from the H.P. diagram, as in Fig. 215, the high-pressure cylinder is decreased in volume in the ratio $\frac{ae}{BC}$ and the cost of the engine is consequently reduced. On account of this saving, and because the expansion should not be to a pressure lower than that which is sufficient to overcome the engine friction, most compound engines are operated with the drop *de* at release in the high-pressure cylinder.

Hence, only that case will be considered in the **analytical method** which follows:

It will be assumed that the expansion is hyperbolic, that the receiver volume is infinite, and that the clearance volumes are zero.

(d) In Fig. 215, let

p_1 = Initial pressure (lbs. sq. in.);

p_2 = L.P. back pressure (lbs. sq. in.);

p_R = Receiver pressure (lbs. sq. in.);

p_D = Release pressure in low-pressure cylinder;

R = Cylinder ratio = (vol. low-pressure cylinder) ÷ (vol. high-pressure cylinder) = V_L/V_H
= (area low-pressure piston) ÷ (area high-pressure piston) when the piston strokes are equal, as they usually are.

r_T = Total ratio of expansion = $\frac{p_1}{p_D} = \frac{V_L}{V_1}$;

r_H = Ratio of expansion in high-pressure cylinder = $\frac{V_H}{V_1}$;

r_L = Ratio of expansion in the low-pressure cylinder = $\frac{V_L}{V_C}$;

p_{mH} = M.e.p. of the steam in high-pressure cylinder (pounds square inch);

p_{mL} = M.e.p. of the steam in low-pressure cylinder (pounds square inch);

p_{mR} = Total m.e.p. "referred" to the low-pressure cylinder (pounds square inch).

Since

$$r_H = \frac{V_H}{V_1} = \frac{V_L}{V_1} \times \frac{V_H}{V_L} = \left(\frac{V_L}{V_1}\right) \div \left(\frac{V_L}{V_H}\right),$$

and since

$$\frac{V_L}{V_1} = r_T \text{ and } \frac{V_L}{V_H} = R,$$

it is evident that the *ratio of expansion in the high-pressure cylinder* is

$$r_H = r_T \div R. \quad . \quad . \quad . \quad . \quad . \quad . \quad (290)$$

(e) As the L.P. piston is R times as large as the H.P. piston (the strokes being assumed equal), the intensity of pressure on the L.P. piston that would do work equal that due to the H.P. mean effective pressure is evidently p_{mH}/R. Then if the high-

pressure cylinder is to do $1/n$ th of the total work, it must follow that the H.P. m.e.p. referred to the L.P. piston will be equal to $\frac{p_{mR}}{n}$, hence

$$\frac{p_{mH}}{R} = \frac{p_{mR}}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (291)$$

Now, from Eq. (278),

$$p_{mH} = p_1\left(\frac{1 + \log_e r_H}{r_H}\right) - p_R, \quad . \quad . \quad . \quad . \quad . \quad (292)$$

and

$$p_{mR} = \left\{ p_1\left(\frac{1 + \log_e r_T}{r_T}\right) - p_2 \right\} \times K, \quad . \quad . \quad (293)$$

in which K is a factor introduced to correct for the loss due to the omission of the toe of the H.P.-diagram. It ranges from 0.9 to 1.0, the latter value being for the complete expansion in the high-pressure cylinder.

Substituting for p_{mH} and p_{mR} in Eq. (291) and solving for p_R, results in the following expression for the *receiver pressure* which will give the desired distribution of work:

$$p_R = p_1\left[\left(\frac{1 + \log_e r_H}{r_H}\right) - \frac{KR}{n}\left\{\left(\frac{1 + \log_e r_T}{r_T}\right) - \frac{p_2}{p_1}\right\}\right]. \quad (294)$$

(**f**) With p_R known the *ratio of expansion in the low-pressure cylinder* can then be found. Since $r_L = \frac{V_L}{V_C} = \frac{p_R}{p_D}$ (see Fig. 215) and since $p_D = \frac{p_1}{r_T}$, it follows that

$$r_L = \left(\frac{p_R}{p_1}\right) r_T. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (295)$$

(**g**) This analytical method not only applies to two-stage compound engines but also to **multiple-expansion engines** having any number of expansion cylinders. Thus, if the work is equally distributed among x cylinders (for example, $x = 3$ in Fig. 216), the work in the first cylinder is $1/x$ th of the total. Then the *pressure* (p_{R_1}) *in the first receiver* can be found from Eq. (294), with x substituted for n.

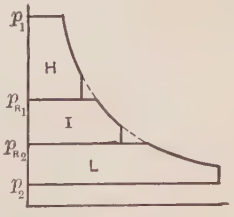


Fig. 216.

The second cylinder receives steam at this same receiver pres-

sure (p_{R_1}); and this cylinder and the succeeding ones can be considered as constituting another engine with initial pressure equal to p_{R_1} and with $(x - 1)$ cylinders. This engine will do $\frac{(x-1)}{x}$ parts of the work of the whole engine, and this second cylinder (considered now as a high-pressure cylinder) will do $\frac{1}{(x-1)}$th of this work. Then the *pressure* (p_{R_2}) *in the second receiver* can be found by again using Eq. (294) with $(x - 1)$ substituted for n and by making such other changes as will be apparent. Pressures in succeeding receivers (if any) can be found in like manner.

(h) In a **triple-expansion engine,** after the ratio R of low-pressure cylinder to high-pressure and ratio R_{IH} of I.P. to H.P. have been selected, it is evident (since $V_L = V_H R$ and $V_I = V_H R_{IH}$) that the *cylinder ratio* R_{LI} *of L.P. to I.P.* is

$$R_{LI} = \frac{V_L}{V_I} = \frac{R}{R_{IH}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (296)$$

(i) Following (f) of this section, the *ratio of expansion in the low-pressure cylinder* is

$$r_L = \left(\frac{p_{R_1}}{p_1}\right) r_T. \quad . \quad . \quad . \quad . \quad . \quad . \quad (297)$$

Also, by analogy, $r_L = \left(\frac{p_{R_2}}{p_{R_1}}\right) r_{T_2}$, in which r_{T_2} is the total expansion in the intermediate-pressure and the low-pressure cylinders combined. After r_L is known, r_{T_2} can be computed from $r_{T_2} = r_L \left(\frac{p_{R_1}}{p_{R_2}}\right)$.

Then by comparison with Eq. (290) it is seen that the *ratio of expansion in the intermediate-pressure cylinder* is

$$r_I = r_{T_2} \div R_{IL} = \frac{r_L}{R_{LI}} \left(\frac{p_{R_1}}{p_{R_2}}\right). \quad . \quad . \quad . \quad . \quad (298)$$

The ratios of expansion in a quadruple-expansion engine would be determined in a similar manner.

170. Cylinder and Expansion Ratios Used in Multiple-Expansion Engines. (a) In general the greater the total range of pressures in the engine the larger should be the cylinder ratio and the expansion ratio. Thus high-pressure engines have

larger ratios than low-pressure engines, and those condensing have greater ratios than those which operate noncondensing. Practice varies widely and only the average values can be given here.

(b) Modern *compound engines* usually operate with steam pressures between 125 pounds and 150 pounds gauge. In many instances, however, much higher and lower values have been used. Stationary engines of this type usually have cut-offs in the high-pressure cylinders between 0.25 and 0.4 of the stroke under normal load. With late cut-off a smaller engine can be used for a given power than with early cut-off; but the consequent saving in "first cost" of engine may be more than balanced by loss in efficiency and greater cost of operation. Cylinder ratios customarily used are about as follows:

CYLINDER RATIOS FOR COMPOUND ENGINES.

Cylinder ratio	$2\frac{1}{2}$	$3\frac{1}{4}$	4	$4\frac{1}{2}$
Gauge pressure, noncondensing	100	120	...	...
Gauge pressure, condensing	...	100	120	150

Dividing the cylinder ratio by the H.P. cut-off fraction (0.25 to 0.4) gives the total ratio of expansion. What the best cylinder and expansion ratios are, is still under discussion. Some advocate cylinder ratios even as large as 6 or 7 and remarkable economies have been obtained with such.*

(c) The ratio of expansion is sometimes fixed by first assuming the pressure drop at release. If this drop is added to the L.P. exhaust pressure, the pressure (p_D in Fig. 215) at release is obtained. Then, considering the expansion to be hyperbolic, the *total ratio of expansion* on the conventional diagram is

$$r_T = \frac{p_1}{p_D}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (299)$$

which is approximated more or less closely in the actual case. If the expansion ratio (r_H) in the high-pressure cylinder is then selected, the *cylinder ratio* is

$$R = \frac{r_T}{r_H} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (300)$$

* $r_T = 6.25$ Cross Compound Corliss. *Am. Electrician*, June, 1903.
$r_T = 7.3$ Fleming Four-valve. Trans. A. S. M. E., Vol. XXV, page 212.
$r_T = 6.4$ Tandem Compound Corliss — Barrus' Engine Tests, page 185.
$r_T = 6.2$ Edison Waterside Station, New York. *Power*, July, 1904, page 424.
Also see papers in Trans. A. S. M. E.

After the receiver pressure, which will give the proper distribution of work between the cylinders, has been determined, the drop in pressure at H.P. release should be checked.

(**d**) Modern *triple-expansion engines* usually operate with steam pressures from 150 pounds to 180 pounds gauge or even higher. The pressure at L.P. release in condensing marine engines is commonly about 15 pounds per square inch absolute under normal load, and in stationary engines it is about half this value. As before, the total expansion ratio (r_T) can be found approximately by dividing the initial pressure by the L.P. release pressure (considering the expansion to be hyperbolic); or it can be obtained from economy curves of similar engines operating under similar conditions, when ratios have been used as abscissas.

The H.P. cut-off in marine engines is usually from 0.55 to 0.7 of the stroke and in stationary engines is much earlier. The H.P. expansion ratio (r_H) is the reciprocal of this cut-off ratio (neglecting clearance). With r_T and r_H known, the *volume ratio* of high-pressure to low-pressure cylinder is $R = \frac{r_T}{r_H}$. If the strokes are equal, as is almost invariably the case, the *ratio of piston areas* will be the same as the volume ratio.

If the conventional diagrams of the various cylinders have sharp toes, the work will be equally distributed among the cylinders if the cylinder volumes (or piston areas) are such that $\frac{H}{I} = \frac{I}{L}$ (in which the letters refer to the high-, intermediate, and low-pressure cylinder volumes, or areas). In such a case the intermediate cylinder volume (or piston area) is found from

$$I = \sqrt{H \times L}.$$

In the actual case, because of departure of the real indicator diagrams from the theoretical and because of cylinder condensation, cushion steam, etc., the intermediate-pressure cylinder is made a little smaller than this equation would give. Seaton* states that in marine practice the intermediate cylinder volume (or piston area) is about as given by the following equation:

$$I = \frac{\sqrt{H \times L}}{1.1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (301)$$

* Seaton's "Manual of Marine Engineering"; or Seaton and Rounthwaite's "Pocket Book of Marine Engineering."

Marine triple-expansion engines are proportioned about as follows:

Initial pressure, abs.	165	175	195
Ratio I to H	2.33	2.4	2.54
Ratio L to H	6.6	7.0	7.8
Total expansion ratio	11.	11.7	13.

(e) *Quadruple-expansion engines* usually operate with pressures from 175 to 225 pounds gauge. The L.P. terminal pressures and H.P. cut-off percentages are about the same as for triple-expansion engines. Thus the total expansion ratios are somewhat larger than in the latter engines. If the ratios of adjacent cylinders are made equal, then

$$\frac{I_1}{H} = \frac{I_2}{I_1} = \frac{L}{I_2} = R_x,$$

in which I_1 and I_2 refer to the first and second intermediate cylinders. From which it follows that

$$I_1 = R_x H \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (302)$$

$$I_2 = R_x I_1 = R_x^2 H \quad . \quad . \quad . \quad . \quad . \quad . \quad (303)$$

$$L = R_x I_2 = R_x^3 H \quad . \quad . \quad . \quad . \quad . \quad . \quad (304)$$

Hence *the ratio of adjacent cylinders* (assuming $\frac{L}{H}$ known) is

$$R_x = \sqrt[3]{\frac{L}{H}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (305)$$

or the ratio of low-pressure to high-pressure cylinder (assuming R_x known) is

$$R = \frac{L}{H} = R_x^3. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (306)$$

After R_x, H, and L are known, I_1 and I_2 follow from Eqs. (302), (303). These values of I_1 and I_2 should be reduced somewhat, for the same reasons that were given in the case of the triple-expansion engine.

In quadruple-expansion marine engines the cylinders are about in the following proportions: — 1 : 1.8 : 3.6 : 7.8. A study * of 14 different quadruple-expansion engines, with pressures about

* H. H. Suplee. Trans. A. S. M. E., Vol. X, page 583.

180 pounds per square inch, showed the average cylinder proportions to be 1 : 2 : 3.78 : 7.70; or nearly 1 : 2 : 4 : 8.*

171. The Theoretical Indicator Diagram of Multiple-Expansion Engines with Clearance. In the foregoing discussion clearance was neglected. If clearance is considered, the results will be changed somewhat. In such cases the analytical method is a little complicated, hence the graphical method is generally the best one to use. This method needs no explanation. In the theoretical cards of a compound engine with clearance, as shown in Fig. 217,

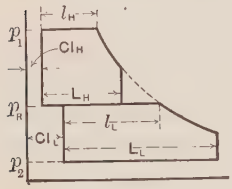


Fig. 217.

the total ratio of expansion is

$$r_T = \frac{L_L + Cl_L}{l_H + Cl_H},$$

the H.P. ratio of expansion is

$$r_H = \frac{L_H + Cl_H}{l_H + Cl_H},$$

the cylinder ratio is

$$R = \frac{L_L}{L_H},$$

and the H.P. and L.P. cut-off ratios are respectively $\frac{l_H}{L_H}$ and $\frac{l_L}{L_L}$.

172. Effects of Changing the Cut-offs in the Respective Cylinders of Multiple-Expansion Engines. (a) In "regulating" the engine to make the power output equal to the demand, the steam distribution to the cylinders can be varied in several ways.

(b) It has already been shown that *the effect of making the L.P. cut-off occur later* in the stroke (other things remaining the same) is (1) to lower the receiver pressure, (2) to increase the H.P. work, (3) to decrease the L.P. work; and *vice versa*. But (4) such change does not affect the total work of the engine if the toes of the diagrams are not lost, hence the engine cannot be regulated by changing merely the L.P. cut-off.

(c) If the **L.P. cut-off is fixed**, and the H.P. cut-off is made to

* For data relating to multiple-expansion marine engines, see Seaton's "Manual of Marine Engineering," Robertson's "Translation of Bauer's Marine Engines and Boilers." For all types of multiple-expansion engines, see Heck's "The Steam Engine," Vol. II, pages 506–9, and Gebhardt's "Steam Power Plant Engineering."

occur later, there results (1) an increase in the receiver pressure (Fig. 218), (2) a greater increase in the L.P. work than in the H.P. work. Making H.P. cut-off occur earlier produces the reverse effects. Compound engines can be regulated by having an automatic governor control only the cut-off in the high-pressure cylinder. But in such case, if there is much change in the load on the engine, the L.P. cut-off should be adjusted by hand to equalize the distribution of the load between the cylinders.

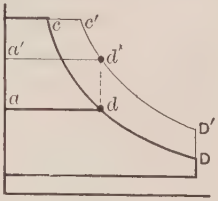


Fig. 218.

(d) If the initial, receiver, and exhaust pressure lines on a PV-diagram for a compound engine are extended from one hyperbolic expansion line to another, as from cD to $c'D'$ in Fig. 219 (a), it will be found (1) that the expansion ratios in the cylinders

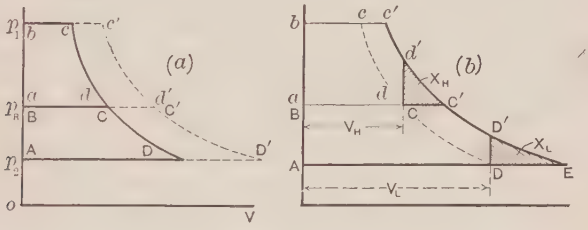


Fig. 219.

remain unchanged; and that, in consequence, (2) the proportionate distribution of work between the cylinders also remains the same.

In Fig. 219 (b) it is seen that the high and low cylinder volumes (V_H and V_L) are such that the expansion lines cd and CD in the two cylinders are complete and continuous. If the cylinder volumes are related thus, and if the cut-offs are advanced proportionately (so that $c'd'$ and $C'D'$ in Fig. 219 (b) are on the same hyperbola), the distribution of work can be shown to be in the same proportion as in the case of complete expansion just discussed; and further (from this), that (3) the toe areas (X_H and X_L) lost will be in this same proportion. These same statements are also true in case the cut-offs are decreased proportionately as in Fig. 220. In this figure, however, it is seen that the diagrams have " loops " X_H and X_L, which represent nega-

tive work. Evidently the cut-off should not be earlier than c, if good economy is important.

With such arrangement the automatic governor can be made to change the cut-off equally in the two cylinders and the proper balance of work will be always automatically maintained. The tandem compound engine in Fig. 107 is an example of this case.

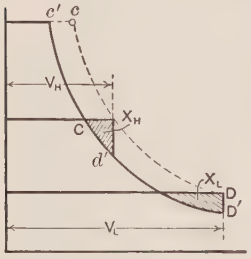


Fig. 220.

If the L.P. toe loss is greater than the similar H.P. loss, it can be shown that to maintain the same relative balance of power between the cylinders, the L.P. cut-off must vary more rapidly than the H.P. cut-off; thus as the power is increased the receiver pressure must be raised.

(e) If the cut-offs (or expansion ratios) in the two cylinders remain constant, the power of the engine may be decreased by *throttling* the steam, and in this case the distribution of power between the cylinders remains in substantially the same proportion. That this is true may be seen from inspection of Eq. (294), in which p_R is seen to be practically proportional to p_1 (since all other quantities are constants in this case, except the ratio p_2/p_1, which is so small a quantity that its change is negligible). This shows that the effect of throttling is substantially equivalent to changing the pressure scale of the diagram.

(f) Because of the effect of clearance, "wire drawing," cylinder condensation, etc., the real diagrams differ greatly from the theoretical ones, hence the conclusions just given can be used only in a very general sense in actual cases.

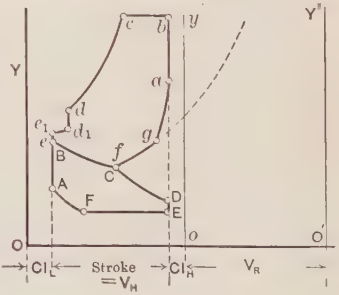


Fig. 221.

173. Theoretical PV-Diagrams of a Tandem Compound Engine with Receiver of Finite Volume, and having Clearance. (a) Fig. 221 shows the PV-diagrams for a tandem compound engine which has clearance volume and finite receiver volume. The

abscissas of both the H.P. and L.P. diagrams are the strokes (same for both cylinders). OY is the line of absolute zero for volumes in the low-pressure cylinder, and oy is the similar line for the high-pressure cylinder. In the latter cylinder abc is the admission line, cd is the expansion line (with respect to axes oy and oO), dd_1 is the drop in pressure when the H.P. steam is released to the receiver, d_1e_1 is the period when the high-pressure cylinder is exhausting into the receiver alone, and e_1ef is the period during which the high-pressure cylinder is exhausting into both the receiver and the low-pressure cylinder; fg shows the period when the high-pressure cylinder is exhausting into the receiver, after cut-off has taken place (at C) in the low-pressure cylinder; and ga is the compression into the H.P. clearance space (and is therefore asymptotic to oy). Evidently if $O'Y'$ is drawn to the right of oy at a distance (V_R) equal to the receiver volume (measured to the same scale that is used for the H.P. volumes), fg will be a hyperbola with axes $O'Y'$ and $O'O$. During the period ef of the H.P. exhaust the low-pressure cylinder is receiving steam along the coincident line BC. After L.P. cut-off at C, the steam in the low-pressure cylinder expands according to CD, is exhausted along DEF, compressed along FA, and admitted along ABC from the high-pressure cylinder and from the receiver. Evidently CD and FA are hyperbolas with respect to axes Oo and OY.

(**b**) These diagrams can also be constructed by the method given in the next section.

174. Theoretical PV-Diagrams of a Cross Compound Engine with Receiver of Finite Volume, and having Clearance. (**a**) In Fig. 222 (*a*) the H.P. and L.P. diagrams of opposite strokes are shown with true volumes as abscissas, and with the clearance and receiver volumes in proper proportion and relation for a single-acting cross compound engine **with L.P. cut-off less than one-half stroke.** It will be seen that the arrangement of diagrams is similar to that in Fig. 212 (*d*), but with clearance and receiver volumes added.

If the points in the stroke at which the valve events occur are known, the lines $abcd$ and EFA are easily drawn, but the points on the H.P. exhaust line and L.P. admission line are harder to find. The method of determining these will now be considered.

(b) It will be convenient to have an auxiliary diagram, such as Fig. 222 (*b*), called a **steam-distribution chart,** which will

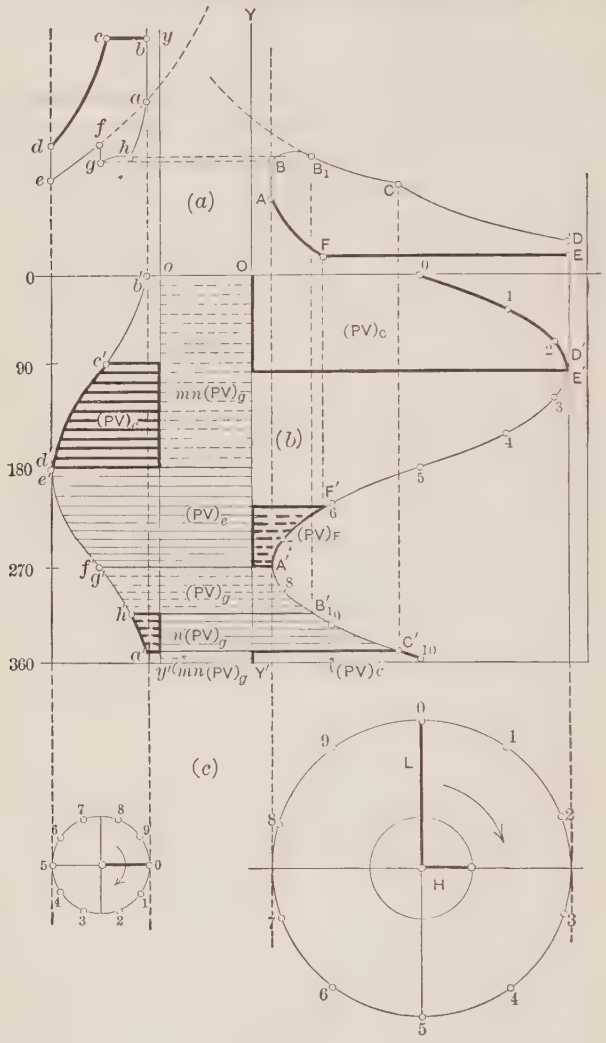


Fig. 222.

show for each crank angle (ordinate) the volumes (abscissas) of steam in both the cylinders and in the receiver. If the motion of the piston is harmonic (as it is approximately), the curves of

volumes displaced by the pistons are of course sinusoids, and can be easily constructed in the manner shown in the lower part of the figure. In the case under consideration, as the cranks are at right angles these sinusoids must differ in phase by 90°. The clearance lines (oy' and OY') are added to the chart; thus the distance from a point on a sinusoid to the clearance line gives the volume of steam in the cylinder for the corresponding crank angle.

The percentages of stroke for all " valve events " are supposed to be known, thus the abscissas of all events can be laid off on the PV-diagrams in Fig. 222 (*a*). Lines *abcd* and *EFA* can be drawn at once, and *efgha* and BB_1C can be drawn tentatively to show roughly the general shape of the diagrams. The exact lines will be found later. Then on the sinusoids, in Fig. 222 (*b*), the points for the valve events can be found by projecting downward from the PV-diagrams, or may be located more accurately by using the crank angles corresponding to the valve events. The points thus found are lettered the same as the corresponding points on the PV-diagrams, but are primed.

(c) From b' to c' in Fig. 222 (*b*) is H.P. admission, and from c' to d' is H.P. expansion, with volumes varying according to the heavy abscissas to the right of the sinusoid between these points. The product PV is constant during this expansion (and its value can be found since P_c and V_c are known), hence the " **PV-quantity** " $(PV)_c$ may be taken as representing the whole process of expansion. Evidently the following broad statement can be made:

General Proposition A: *Between valve events* (not necessarily in the same cylinders) *controlling the weight of steam involved, the " PV-quantity " is constant;* and when its value is known the expansion curve can be constructed. Thus, in this instance, dividing the PV-quantity $(PV)_c$ by different values of V gives the pressures to be used in plotting the expansion curves *cd*.

(d) At d (and d') the steam with PV-quantity equal to $(PV)_c$ is released from the high-pressure cylinder and mixes with the receiver steam which has a PV-value equal to $mn(PV)_g$, in which m and n are unknown coefficients, the value of which will be determined later. In such cases the following assumption is made:

General Proposition B: The *PV-quantity resulting from a mixture* is

$$[PV]_x = \Sigma\,(PV). \quad . \quad . \quad . \quad . \quad . \quad . \quad (307)$$

Thus, after point e is passed

$$[PV]_e = (PV)_c + mn\,[PV]_g, \quad . \quad . \quad . \quad (308)$$

from which $[PV]_e$ can be found when $mn\,[PV]_g$ has been determined, since $(PV)_c$ is already known.

(e) The L.P. compression occurs from F to A (and F' to A') with PV-quantity constant and equal to $(PV)_F$, — the value of which can be easily found, since P_F and V_F, are given, — and with volumes varying as shown by the heavy dotted abscissas to the left of sinusoidal arc $F'A'$. At A (and at A' and f') this L.P. cushion steam mixes with that in the receiver and high-pressure cylinder; hence the PV-value of the mixture is, from Proposition B,

$$[PV]_g = [PV]_e + (PV)_F. \quad . \quad . \quad . \quad . \quad (309)$$

Thus during phase gh and BB_1 the pressures may be found by dividing $[PV]_g$ by the volumes which are shown by the dotted abscissas between arcs $g'h'$ and $A'B_1'$.

(f) After the H.P. exhaust valve has closed at h there remain in the receiver and low-pressure cylinder n parts of the steam that has been represented by $[PV]_g$, and the rest, $(1 - n)$ parts, is used for compression in the high-pressure cylinder. Between B_1 and C (and B_1' and C') the PV-quantity of the steam in the low-pressure cylinder and receiver is $n\,[PV]_g$ in accordance with the following assumption:

General Proposition C: *If a weight of steam, having a certain PV-quantity, is divided without change in pressure, the PV-quantity of the part is the same fraction of the original PV-quantity that its weight is of the original weight.* For example, if one-half the steam involved is left in the cylinder and receiver, when the H.P. exhaust closure occurs at h or B_1 (h' or B_1'), then $n = \frac{1}{2}$, and the PV-quantity of this remaining steam has the value $\frac{1}{2}\,[PV]_g$. Thus, between points B_1 and C the PV-value is $n\,[PV]_g$ and the volumes are shown by the abscissas to the left of the sinusoidal arc between points B_1' and C'.

After the L.P. valve has cut off at C (and C') there are left in the receiver m parts of the steam which was represented by $n\,[PV]_g$; hence, this receiver steam has a PV-value of $mn\,[PV]_g$,

which continues constant until point a in the next cycle is reached.

(g) In the simultaneous equations (308) and (309) all quantities are either known or can be determinable directly, except the bracketed quantities $[PV]_e$ and $[PV]_g$; but these latter can be found by elimination. When these are known, the PV-diagrams can easily be completed.

(h) If the engine is **double-acting,** and if it has equal PV-diagrams at both ends of the cylinders, the solution of only one end is necessary. But if the diagrams are not equal, it is necessary to draw the steam-distribution chart for both ends of the cylinders. Then there will be four unknown PV-quantities, but there will be the following four simultaneous equations, from which the unknowns can be determined:

$$[PV]_e = (PV)_c + m'n'\,[PV]'_g \quad . \quad . \quad . \quad (310)$$
$$[PV]_g = [PV]_e + (PV)_F \quad . \quad . \quad . \quad . \quad (311)$$
$$[PV]'_e = (PV)'_c + mn\,[PV]_g \quad . \quad . \quad . \quad (312)$$
$$[PV]'_g = [PV]'_e + (PV)'_F \quad . \quad . \quad . \quad . \quad (313)$$

in which the primed quantities are those for the cylinder ends not considered in the previous discussion.

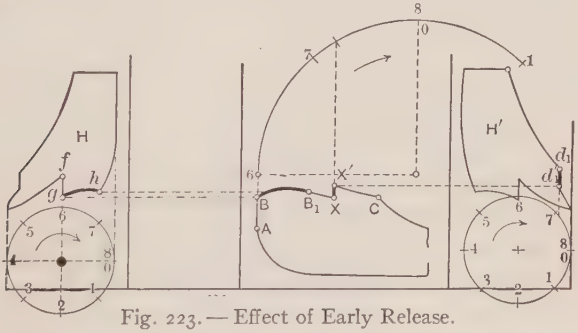


Fig. 223. — Effect of Early Release.

(i) In the foregoing it has been assumed that release and admission occur at the ends of the stroke. If the engine is *double*-acting and if the steam is **released before the end of the stroke** in the high-pressure cylinder, the L.P. admission line will suddenly rise, in case the L.P. cut-off has not already occurred; for this release suddenly increases the steam pressure in the receiver from which steam is still being supplied to one end of the low-pressure cylinder. This is shown in Fig. 223, from which it

is seen that, when the steam is released at d_1, there is a drop of pressure in the high-pressure cylinder accompanied by a simultaneous rise at X in the low-pressure cylinder, until the pressures at X' and d'_1 are equal. This case can be analyzed by the method already given.

(j) **The case with cut-off later than half-stroke** is somewhat similar to that discussed in (i) of this section and is illustrated in Fig. 224. Even if the H.P. release occurs at the end of the stroke (at d_1), there will be the sudden rise XX' on the L.P. admission line, as the low-pressure cylinder has not previously

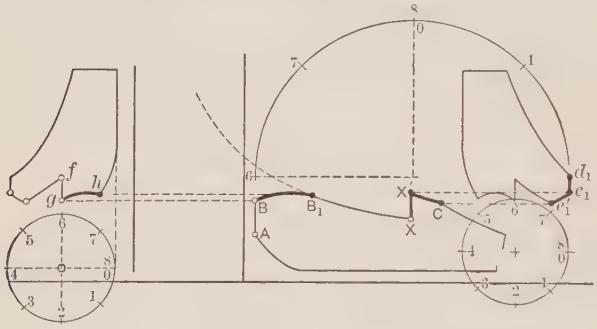


Fig. 224. — Low-pressure Cut-off later than Half-stroke.

been cut off from the receiver. From X' to cut-off at C the low-pressure cylinder continues to receive steam from the receiver, while simultaneously the high-pressure cylinder is discharging steam into the receiver according to line e_1e_1'. Evidently the pressures at points e_1 and X' are equal; and the same is true of points e_1' and C. This case can be analyzed with the aid of a steam-distribution chart, in the same manner as that which has just been discussed in connection with the other cases.

175. Theoretical PV-Diagrams of Multiple-Expansion Engines with Finite Receiver and Clearance Volumes, with Any Number of Cylinders and with Any Angles between Cranks (General Case). The methods just discussed in connection with the construction of PV-diagrams for compound engines can be extended to this perfectly general case. It is assumed that the initial and exhaust pressures are known, together with the volume ratios of cylinders, receivers, and clearances, and that the

percentages of stroke (or crank angles) of the various valve events are given. The procedure is as follows:

(1) Draw the cylinder, clearance, and receiver volumes in proper relative positions on the PV-diagrams.

(2) Sketch as much of the H.P. and L.P. PV-diagrams as can be done initially.

(3) Draw the sinusoids on the steam-distribution chart in a proper phase relation (considering the crank angles and "sequence" of cranks); locate the valve events; and by a system of section lining show the volumes connected between events (remembering that these volumes are not necessarily confined to those in one cylinder).

(4) On the distribution chart:— (*a*) give the PV-quantities initially known, such as $(PV)_c$ and $(PV)_F$ in the previous cases; (*b*) in accordance with General Proposition C state the PV-quantities resulting from a separation of volumes (when not accompanied by change in pressure) as fractional parts of the quantity which is divided, as $mn\,(PV)_g$; and (*c*) in accordance with General Proposition B, write equations for the PV-quantities resulting from mixtures.

(5) Obtain the values of the fractional coefficients, m, n, etc.

(6) Find the unknown PV-quantities by solving the simultaneous equations, of which there should be the same number as there are unknowns.

(7) Complete the construction of the PV-diagram, which can be done now that the PV-quantities are all known.

The heavy lines in Fig. 225 show the PV-diagrams for a triple-expansion engine. The individual diagrams were first obtained in the manner just outlined and then were combined with respect to a common axis of volumes as shown in this figure.

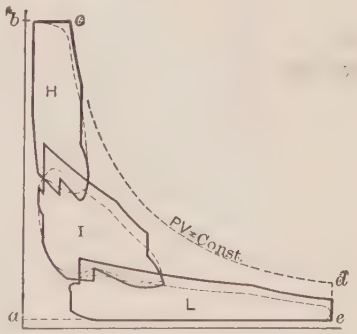


Fig. 225.

176. The Actual Combined Indicator Diagrams of Multiple-Expansion Engines. (a) In Fig. 225 it is seen that the **theoretical PV-diagrams** (in heavy lines) overlap, that their expansion

lines do not fall on the same hyperbola, and that the sum of their areas is much less than that of the simple diagram *abcde*. The overlapping parts of the diagrams do not occur simultaneously. The lack of continuity of the expansion lines is largely due to the difference in the amounts of cushion steam in the various cylinders; it is also due to the sloping and irregularity of the I.P. and L.P. admission lines, and to earliness of the I.P. and L.P. cut-offs compared with that in the high-pressure cylinder. The ratio of the sum of areas *H*, *I*, and *L* to area *abcde* is the theoretical diagram factor in this case, and it is evidently much less than unity.

(**b**) The actual indicator diagrams depart considerably from the theoretical. This is partly because of wire drawing during flow of steam through valves, receivers, and piping, partly because of condensation or reëvaporation in cylinders, receiver, and piping, partly from radiation and similar losses, partly because the real expansion line is not hyperbolic, and may also be partly due to the withdrawal of the condensate collecting in " separating " receivers. In Fig. 225 the *probable diagrams* are shown dotted.

(**c**) Given the *actual indicator* cards obtained from the en-

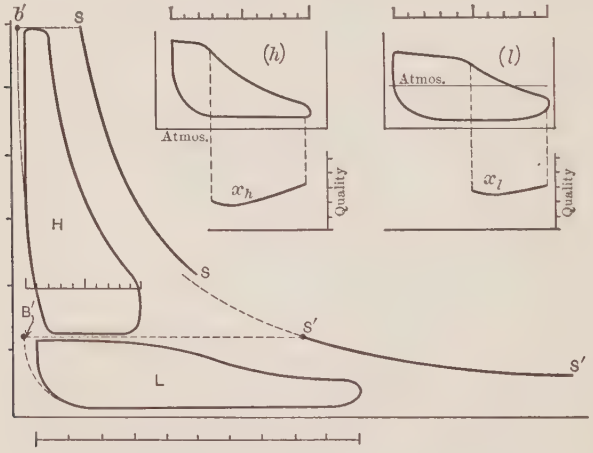


Fig. 226.

gine, as *h* and *l* in Fig. 226, they can be readily " combined," as shown by *H* and *L*, if the cylinder and clearance volumes are known.

The areas of diagrams H and L can then be found and the "referred m.e.p." determined in the usual manner.

After this, the actual diagram factor can be obtained by getting the ratio of these quantities to the area of the conventional diagram.*

(d) On Fig. 226 the **saturation curves,** SS and $S'S'$, have been drawn. As the weights of cushion steam in the two cylinders are not the same, and because the condensate has been removed from the receiver in this case, there are unequal weights of steam in the two cylinders during the respective expansions, consequently saturation line $S'S'$ lies to the left of SS.

Fig. 226 also shows the *quality curves* x_h and x_l, which are obtained, after the saturation lines have been drawn, by the same method that was described for simple engines.

176A. Clayton's Analysis of Expansion Lines. (a) By replotting indicator cards on logarithmic coördinates Clayton† has determined the expansion coefficients n for many engines. He found that such expansion lines were substantially straight (except when modified by leakage or faulty indicator practice which, if present, were revealed by the curvature); that the points of cut-off and other valve events could be accurately determined; and that, as in the ideal case (see (d), page 207), there appeared to be a definite relation between the quality (x_c) at cut-off and the exponent of expansion (n) for each type of engine and condition of operation.

For a Corliss non-condensing engine, without leakage, he found

$$x_c = 1.245\, n - 0.576. \quad . \quad . \quad . \quad . \quad . \quad (313a)$$

There was some variation with change of speed and pressure; but cylinder size and point of cut-off had little influence. Having determined n for the engine x_c can be computed, after which the water rate can be found.

With the logarithmic coördinates the clearance volume can be determined quite closely, when there are no abnormal disturbances, by finding the origin which will give a straight line.

* There are several different kinds of diagram factors, each of which may be used to best advantage for some particular purpose. When the engine is considered by itself, the definition previously used in the text is the one most commonly given. The A. S. M. E. Report of Committee on Standardizing Engine Tests defines the card factor in such manner as to include the cylinder-feed losses between engine and boiler. See Trans. A. S. M. E., Vol. XXIV, page 751.

† Trans. A. S. M. E. 34, p. 17. Bulletin 58 and 65, U. of Ill. Exp. Sta.

CHAPTER XXI.

PERFORMANCE OF STEAM ENGINES.

177. Steam Consumption. (a) Steam engines are governed by (1) throttling the steam, (2) by varying the cut-off, and (3) by combining (1) and (2).

When the engine is **governed by throttling** (the cut-off remaining constant), the available energy ΔE per pound of steam theoretically decreases as the pressure is reduced. This is shown in the Mollier diagram, Fig. 227. Starting with initial pressure p_1, the associated heat ΔQ_1, and back pressure p_b, the available energy is ΔE_1. In throttling to pressure p_2 the associated heat remains unchanged, but the available energy per pound is reduced to ΔE_2, and consequently more steam, in the ratio $\left(\frac{\Delta E_1}{\Delta E_2}\right)$, must be used to develop one i.h.p.-hour. Evidently the actual throttling engine gives the best economy only under maximum load, hence the water-rate curve will resemble *cd* in Fig. 228.

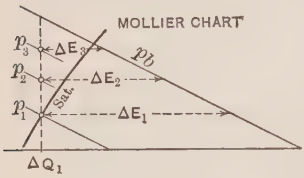


Fig. 227.

It is found that with this type of governing, the curve of total consumption (*ab*) is practically a straight line, and this relation is commonly called **Willans' Law.** When two points on this line, or one point and the slope, are given, the line can at once be drawn. Then dividing ordinates by corresponding abscissas gives the simultaneous values of the water rate, and these values can be used for plotting the water curve.

With greater ratio of expansion, less steam is used for a given output, hence for such cases the curves $a'b'$ and $c'd'$ in Fig. 228 would lie below the others.

(b) When the engine is governed by **varying the cut-off**, the water-rate curve resembles *efg* in Fig. 229, the reasons for which were made clear in Section 125. To this figure has been added

the curve *cd* of Fig. 228, the point *d* of course coinciding with *g*. Thus it is seen that cut-off governing gives better results than throttle governing except at the maximum load.

The product of abscissas by ordinates gives the total steam consumption, plotting which gives the curved line *hij* as the Curve of Total Water Consumption for cut-off governing. Evidently point *i*, where a line drawn from *O* becomes tangent to *hij*, determines the abscissa for the lowest water rate, for that point has the smallest ratio of ordinate to abscissa.

(c) The *y* intercept *Oy* of the T.C. curve represents the weight of steam used when no i.h.p. is being developed. It is the weight which furnishes heat equivalent to the losses from condensation, leakage, and radiation.

Curves similar to Figs. 228 and 229 might have B.t.u. as ordinates; and m.e.p's, cut-offs, ratios of expansion, or d.h.p's may be

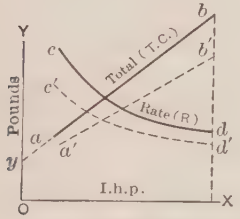


Fig. 228.

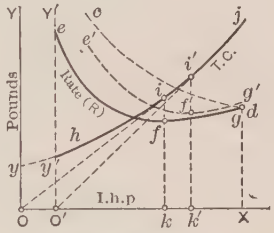


Fig. 229.

used as abscissas. When abscissas are d.h.p., then the *y* intercept represents the consumption due to engine friction in addition to the other losses mentioned in the preceding paragraph.

(d) If, in Fig. 229, *OO'* is the i.h.p. used in overcoming the engine friction, then *O'Y'* is the axis from which the d.h.p. are measured. If the engine friction is assumed constant for all loads (which is not strictly true), the curve *TC* in the figure, with origin at *O'*, gives the total consumption for the d.h.p. developed. The curve *e'f'g'* of water rate per d.h.p.-hour will of course lie above *efg*, and the lowest point *f'* will lie farther from *O* than *f*. Evidently, on the basis of *delivered power*, the best economy in this case occurs when the i.h.p. equals *Ok'* (corresponding to a d.h.p. of *O'k'*), and this should be the power which the engine normally develops (" Normal Power ") if steam economy is of prime importance. This should then be the " rated

power," or power at which the engine is rated to operate normally. When the i.h.p. developed is either more or less than this, the engine has poorer economy.

(e) The **load factor** is the ratio of the actual load to the rated load. There are instantaneous load factors, and average load factors. For best *steam* economy the load factor should be unity; and, since it is better to overload than to underload a steam engine (see Fig. 229), a load factor a certain amount above unity is preferable to one the same amount below. There are, however, other considerations which may make it financially more profitable to rate the engine at output other than that giving best steam economy, and to operate with some load factor other than unity.

In many instances, the average load factor of the power plant as a whole is low, but in such cases it is customary, when possible, to have several engines and to place in operation such a number as will cause those in service to operate under the most economical conditions; that is, the load factors of the individual engines are maintained near unity.

The instantaneous load factor may vary widely, as in a street-railroad power plant, and the fluctuations may be of such rapid character as to prohibit changing the number of engines. In such a case a small average load factor may be unavoidable.

(f) Curves of steam consumption for an engine are useful in determining the best conditions of operation for that particular engine and for comparing it with others that operate under *similar* conditions. When the conditions are widely different the water rates should not be compared directly.

To reduce water rates to a comparable basis, when the difference in conditions of operation is not great, the following corrections may be made:*

0.4 to 0.6 per cent per 1 inch change in vacuum (between 25 and 28 inches).

1 per cent per 8 to 11 degrees of superheat (at from 50 to 100 degrees).

0.1 to 0.2 per cent per pound of initial pressure.

1 per cent per 1 per cent of moisture.

The only **true comparison** is on the basis of B.t.u. per h.p. per unit of time (minute) or on the basis of thermal efficiency

* Moyer's Steam Turbines, page 288: — Wiley & Sons.

TABLE VII. — SUMMARY OF PERFORMANCES OF STEAM ENGINES.

	Number.			Horse Power.	Initial Pressure, Lbs. Gauge.	Back Pressure, Lbs. Abs.	Temperature of Feed Water.	Superheat at Admission. °F.	Pounds of Dry Steam per i.h.p. Hour.‡	B.t.u. per i.h.p. Minute.	Thermal Efficiency.	Cylinder Efficiency.	M.e.p. Referred to L.P. Cyl.	Remarks.
Simple (1×)	1	Single-valve	Range	33-257	80-124	Atm.	.....	.....	26.0-30.6	436-510	8.33-9.75	57.5-65.7	35.0-58.5	
		Noncondensing	Avg.	137	104.9	Atm.	.....	.....	27.03	403	9.20	61.3	44.5	Avg. 7 engines.
	2	Single-valve	Range	33-204	60.3-114	1-3.2	.....	.....	22.2-27.5	410-490	8.67-10.35	39.7-41.3	38.1-41	
		Condensing	Avg.	116	83.3	2.4	.....	.....	25.7	450	9.47	40.5	39.9	Avg. 4 engines.
	3	Four-valve	Range	120-506	91.6-125.4	Atm.	.....	.....	22.24-25.9	374-434	9.80-11.30	66.2 73.5	25.7-48.4	
		Noncondensing	Avg.	264	106.9	Atm.	.....	.....	24.06	404	10.54	70.2	38.6	Avg. 5 engines.
	4	Four-valve	Range	145-613	67-96	1.0-2.9	.....	.....	18.5-22.0	342-397	10.7-12.4	48.0-60.2	30.9-38.2	
		Condensing	Avg.	381	78.2	1.9	.....	.....	19.84	359	11.85	51.3	35.0	Avg. 4 engines.
Comp. (2×)	5	Noncondensing	Range	33-1125	114-210	Atm.	212	.....	17.17-22.3	291-376	14.5-11.2	65.5-83.0	31.56	Cyl. ratios 1:2 to 1:3.8.
		Sat. steam	Avg.	350	151	Atm.	*	.....	20.3	342 3†	12.4	72.3	37.1	Avg. 10 engines.
	6	Condensing	Range	340-7365	100-185	0.8-2.2	100-130	.....	11.20-12.70	220-234	18.1-19.2	63.5-76.5	13.0-27.9	Cyl. ratios 1:3½ to 1:7.3.
		Sat. steam	Avg.	2770	152	1.47	119.3	.....	12.14	223.6	18.8	69.0	21.2	Avg. 8 engines.
	7	Condensing	Range	145-2202	114-163	0.84-2.15	97-130	40-343	8.58-11.80	176.1-223	24.0-19.0	78.5-54	.....	Cyl. ratios 1:2.67 to 1:7.3.
		Superheated steam	Avg.	673	132	1.39	112*	218	10.0	202 2†	**20.98**	69.0	.....	Avg. 8 engines.
(3×)	8	Condensing	Range	404-1823	134-185	0.85-1.8	122-158	.....	10.33-11.33	196-208	20.4-21.63	70.0-76.0	19.5-23.4	Cyl. ratios 1:2.2:5.9 to 1:3½:8.4
		Sat. steam	Avg.	988	152	1.22	145	.....	10.81	202.2	20.98	72.8	21.05	3 pumping, 1 mill engine.
	9	Condensing	Range.	549-2040	147-173	0.79-1.28	95-111	87.2-264	8.97-10.00	188.7-207.3	20.4-22.6	66.7-73.5	.....	Cyl. ratios 1:2.05:3.22 to 1:2.64:5.9
		Superheated steam	Avg.	1517	165	1.00	102*	195	9.65	198.4†	21.4	70.1	.....	Avg. 5 engines.
(4×)	10	Condensing	Range.	712-990	200-243	0.9-1.25	310.8-334.5	.....	12.26-11.25	186-169.3	22.8-25.05	74.2-88.2	35.5	2 engines, regenerative cycle.
		Sat. steam	Avg.	851	221	1.07	322.6	.....	Cyl. ratios 1:1.35:3.9:8.7 and 1:2.331:7.01:14.19					Trans. A. S. M. E. 1900 and 1907.
1×	11	Locomotives, sat. steam		399 & 975	110 & 196	Atm.	.....	.....	24.97-23.4	420-398	10.1-10.65	65.0-5.59	54-75.6	Trans. A. S. M. E., Vol. XIV.
2×	12	Locomotives, sat. steam		495	210	Atm.	212*	.....	18.6	316†	13.4	68.5	55	Loco. tests, 1904, Louisiana Expo.
1×	13	Corliss, jacketed, N. C. sat.		237	103.5	Atm.	.....	.....	21.5	358	11.85	78.0	42.1	Peabody's "Thermodynamics."
1×	14	Corliss, jacketed, cond. sat.		155	103.8	1.2	.....	.....	16.5	302	14.05	53.3	32.0	Peabody's "Thermodynamics."
1×	15	Poppet, jacketed, cond. sat.		262	79	1.36	.....	.....	15.0	275	15.4	63.8	30.2	*Zeit. d. v. D. Ing.*, Aug., 1905.
3×	16	Binary, superheated		211	143	2.2	.....	221	8.6	158.3†	26.8	.....	.....	Journal Franklin Institute, 1902.
2×	17	White automobile, superheated		40	426	Atm.	212	316	11.96	244	17.4	68	.....	Trans. A. S. M. E., Vol. XXVIII.

* Ideal feed-water temperature (temperature of hot well). ‡ Pounds of superheated steam, or weight of saturated steam corrected for moisture.

† Above ideal feed-water temperature.

1 × = simple engine; 2 × = compound; 3 × = triple; 4 × = quadruple expansion

on the d.h.p. The engine having the lowest water rate and the highest cylinder efficiency does not always use the least heat per unit of power, nor have the highest thermal efficiency.

178. Steam-Engine Performance: Data. — (a) The performance of engines is dependent on many things, of which the more important ones are: (1) initial pressure, (2) back pressure (condensing, noncondensing), (3) cut-off, or expansion ratio, (4) number of expansion cylinders, (5) quality, or superheat, (6) use of jackets, (7) use of reheating receivers, (8) speed, and (9) the proportions, size, and arrangement of cylinders, clearance spaces, and passages. These items must be considered in comparing economies.

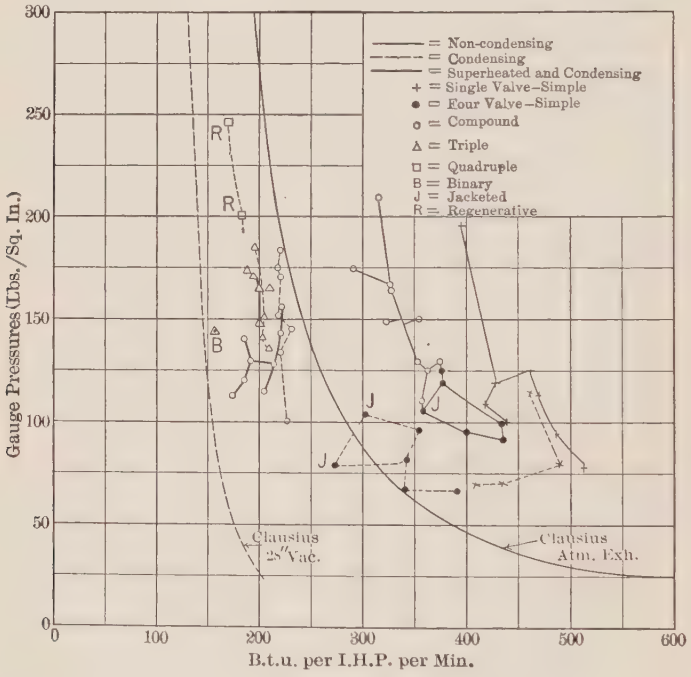


Fig. 230.

Table VII is a brief summary of Gebhardt's more extensive tables,* with a few additions and omissions. It will serve as a

* See Gebhardt's "Steam Power Plant Engineering," pages 296, 306, and 314. Wiley & Son, publishers.

Also Heck's "The Steam Engine," Vol. II, pages 600–652.

rough comparison of some of the best performances that have been obtained with the principal types of engines. The condensing, multiple-expansion engines are in most cases steam-jacketed unless the steam is superheated.

Although the tabulation as here given does not bring out this point, it should be remembered that while the lowest B.t.u. per i.h.p.-min. corresponds to the highest thermal efficiency, it

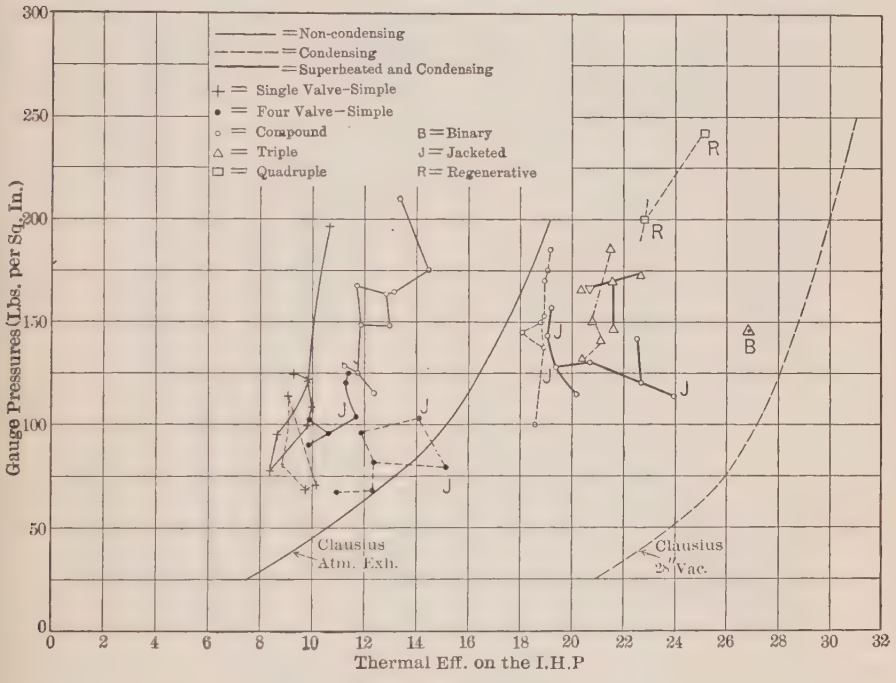


Fig. 231.

does not follow that it is accompanied by low water rate and high cylinder efficiency.

(b) Figs. 230 and 231 show respectively the variations of B.t.u. per i.h.p.-min. and the thermal efficiency on the i.h.p., with initial and exhaust pressure, with superheat, with type of engine, etc. Reference to these figures shows that, as compared with the Clausius cycle with 28-inch vacuum, the losses of various types of real engines are about as follows in the best condensing practice:

Quadruple	20%
Triple	25%
Compound	33%
Simple	50%

(c) Table VIII gives a brief summary of steam-engine efficiencies, including some of the best. Table IX gives the pounds of steam consumed per i.h.p.-hour by ordinary engines which operate under the usual commercial conditions and in which no special provision is made for improving economy — such as superheating, jacketing, etc. Larger engines, of course, give better results than smaller ones.

TABLE VIII. — SUMMARY OF EFFICIENCIES OF STEAM ENGINES.

Carnot cycle efficiency *	10	to	32%
Clausius cycle efficiency *	8	to	28%
Indicated efficiency	40	to	88.2%
Thermal efficiency on i.h.p.	5	to	25.05%
Mechanical efficiency	85	to	97%
Thermal efficiency on d.h.p.	4	to	23.9%
Over-all efficiency	35	to	84%
Heat used per i.h.p.-min.	169.3	to	700 B.t.u.

TABLE IX. — STEAM CONSUMPTION.

Type of Engine.	Lbs. i.h.p.-hour
Simple "high-speed" engines (noncondensing)	28 to 36
Simple Corliss engines (noncondensing)	25 to 28
Compound slide-valve engine (noncondensing)	24 to 26
Compound slide-valve engine (condensing)	15 to 21
Compound Corliss engine (condensing)	14 to 16
Triple-expansion (condensing)	$12\frac{1}{2}$ to 13

* Obtained from Figs. 73 and 75 respectively with the following assumptions: The lower limit of p_1 is assumed at 50 pounds pressure and $t_2 = 212°$ F. The upper limit of p_1 is assumed at 150 pounds pressure and $t_2 = 100°$ F.

CHAPTER XXII.

STEAM TURBINES.

179. Introductory. (a) The earliest steam-driven prime mover recorded in history is **Hero's** steam turbine (about 200 B.C.), which is shown in Fig. 232. It was a "reaction turbine," driven by the repulsive force produced by a jet of steam issuing rearwards as regards the direction of rotation. **Branca's** "impulse turbine"

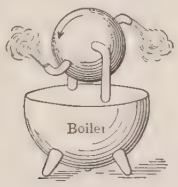


Fig. 232.

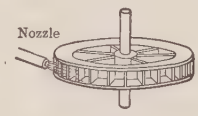


Fig. 233.

(1629), shown in principle in Fig. 233, is the next historical reference to the use of steam in a turbine. The first patents in foreign countries appeared about 1820 and the primary patent in the United States was issued in 1831. Although many steam turbines were invented in the succeeding years, it was not until the latter 80's of the last century that the modern commercially successful types began to be developed.

(b) A steam turbine may be defined as a device in which one or more jets of the working substance moving at high velocity (and therefore possessing kinetic energy) act or react on vanes or buckets on one or more wheels, or drums, in such manner as to cause them to rotate and transmit power by means of the shaft on which they are mounted.

The shaft, the wheels or drums, and their attachments constitute the **"rotor."** The working substance is steam, which may have moisture entrained in it. The velocity of the jet is acquired by the expansion of the steam through a nozzle, or its equivalent, during which process some of the heat energy of the steam is converted into the kinetic energy of the issuing jet.

In "**impulse turbines**" the nozzles are stationary and the jets act on the turbine vanes; in the "**reaction**" type, the nozzles, or their equivalents, are mounted on the rotor, which is driven by the reaction of the jet. In some turbines the rotors are driven by both impulse and reaction.

(**c**) The velocity diagrams used in designing the buckets of the steam turbine are similar in many respects to those used for water turbines. But, despite this resemblance, the problems of design and construction in the former differ greatly from those in the latter. This is principally because, in the steam turbine, (1) the jet velocities are enormously greater (in some cases this velocity exceeds 3600 feet per second, or 41 miles per minute), (2) the bucket velocities are very much higher, (3) the working substance is elastic and tends to expand as fully as the surrounding media will allow, and (4) because the kinetic energy of the jet is obtained from *heat* conveyed by the working substance and not from "hydraulic" head.

(**d**) The steam turbine differs as much from the steam engine as to its mechanism and method of operation as does the water turbine. Although both of these steam-actuated prime movers use the available heat of the steam, the turbine utilizes it in increasing the velocity (kinetic energy) of the jet of working substance, whereas this heat in the steam engine produces certain pressure-volume changes within a cylinder.

(**e**) The *thermodynamic problems* encountered in the steam turbine are centered in the *nozzle*, where (theoretically) all the heat-energy transformations occur. After the jet has issued from the nozzle end the problem becomes a *dynamic* one, namely, to convert the jet's *kinetic energy into power* which can be delivered by the shaft.

The problem of nozzle design and the thermodynamic theory involved will be considered in detail in a later chapter. For present purposes it is only necessary to know that high velocity can be attained at the expense of associated heat and that this transformation occurs entirely within the nozzles or their equivalent.

(**f**) In turbines, there is a certain definite ratio of bucket velocity to jet velocity that will theoretically give the best economy. In practice, however, if the full expansion from initial to final pressure takes place in a *single* set of nozzles, the bucket velocity for

best economy is usually greater than the structure of the rotor will stand, because of the enormous centrifugal force produced. Also, the high rotative speed involved with high bucket speeds usually prohibits the direct connection of the driven machinery to the turbine shaft. Hence, if the expansion occurs in a single set of nozzles, it is usually necessary to use lower bucket velocities than those which would give the highest economy, and also to use gearing of some kind between the turbine and the machine it drives.

In order to obtain lower jet and bucket velocities, most turbines are of the "**multi-stage**" type. Fig. 234 shows diagrammatically an impulse turbine of this type. In such turbines each stage by itself constitutes a simple turbine, in the nozzle of which the steam expands through a small range and therefore acquires relatively low velocity. The stages are usually arranged in series with diaphragms between and with all rotors mounted on the same shaft.

In Fig. 234, the sections of the turbine casing and the diaphragms are shown by crosshatching, and the nozzle and turbine wheel sections are black. Steam enters at the left, expands through the first nozzle (or ring of nozzles) N_1, in which it acquires a relatively low velocity, and discharges against the buckets on the wheel in the first-stage casing, in which the pressure is but little lower than the initial. The steam then expands through the nozzle (or ring of nozzles) N_2, in the diaphragm between the first and second stages, and acts on the buckets of the wheel in the second chamber, where the steam pressure is somewhat lower than it is in the first stage. In similar manner the process is continued in a third stage, and in many instances in from twenty to forty stages, until the exhaust pressure is reached in the last stage.

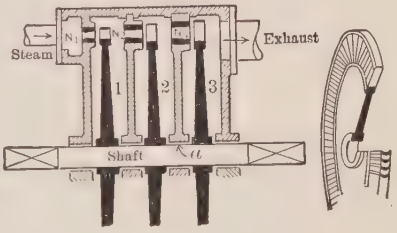


Fig. 234.

The nozzles in all the stages must all deliver the *same weight* of working substance per second. They may be designed to do this with *equal velocities*, in which case the bucket velocities in all stages would be the same and the mean diameters of the

wheels would be equal; or the *jet velocities* may be *varied* and the bucket velocities and wheel diameters be made to correspond.

As the steam traverses the turbine it expands by increments in the successive nozzles, and increases in volume, hence the nozzle areas must increase in like manner through the series, as is illustrated in Fig. 234.

By properly proportioning the cross-sectional areas of the inlet and outlet nozzles of any stage, the designer can fix at any desired value the pressure that will be maintained in that stage.

180. Thermodynamics of the Ideal Steam Turbine. (a) In all types of steam turbine the steam is expanded through nozzles, or their equivalent, and the velocity of the working substance itself is increased by the conversion of some of its own associated heat into available mechanical energy, which appears as the kinetic (velocity) energy of the issuing jet. As the nozzles, or their equivalent, are relatively small, and as the velocity of the steam through them is enormous, there is little opportunity for loss of heat, as such, to the surrounding media, or for the reception of heat, as such; hence, the conversion of heat energy into kinetic energy must in practice be almost strictly adiabatic, and it will be shown in a later chapter that the expansion may be considered equivalent to an isentropic process in the ideal case.

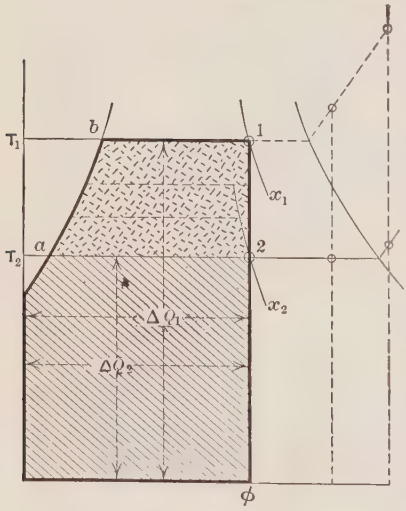


Fig. 235.

(b) Assuming that the steam is initially wet, and that its state is represented by point 1 on the $T\phi$-diagram in Fig. 235, the heat (ΔQ_1) supplied per pound of steam delivered to the turbine is represented by the area bounded by the bold line. Let point 2 represent the state of the working substance after isentropic expansion to the exhaust pressure and temperature. Then the hatched area represents the heat (ΔQ_2)

remaining in the steam at the end of the process. Thus the heat theoretically available for the turbine to deliver as useful work is

$$\Delta E = \Delta Q_1 - \Delta Q_2,$$

and this is shown by the stippled area, *ab*12, which is seen to have the same boundary lines as those of a Clausius cycle, with the same conditions of expansion. The isentropic process from 1 to 2 may occur in one nozzle, converting ΔE into kinetic energy, or it may occur in any number (n) of nozzles in series, each converting part of ΔE, but with cumulative effect equal to that produced by ΔE in the single nozzle. Thus, regardless of the number of stages, it may be said that *the heat energy available for doing work in the steam turbine is equivalent to the ΔE available with the Clausius cycle* having the same expansion line and same weight of steam. The value of ΔE per pound may be computed by the method given on page 173; or it can be obtained from the area on the Tϕ-diagram; or it can be more conveniently found from the Mollier chart (Appendix).

(c) Having determined the number of B.t.u. represented by ΔE, the steam consumption per h.p.-hour, or **water rate**, in the ideal turbine is

$$W_I = \frac{2545}{\Delta E}, \quad \frac{\text{B.T.U.}}{\text{B.T.U./lb.}} \quad . \quad . \quad . \quad (314)$$

and if the turbine drives an electric generator the theoretical water rate per kilowatt-hour is

$$W_{IK} = \frac{1}{0.746} \cdot \frac{2545}{\Delta E} = \frac{3411}{\Delta E} \quad . \quad . \quad . \quad . \quad . \quad (315)$$

(d) The actual turbine of course has a poorer (larger) water rate than the ideal. If W_d is the actual water rate per h.p.-hour delivered by the turbine shaft, and W_{dK} is that per kilowatt-hour delivered by the generator, then the **over-all efficiency of the turbine** (alone) is

$$OEf_d = \frac{W_I}{W_d}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (316)$$

and the **over-all efficiency of turbine and generator** is

$$OEf_k = \frac{W_{IK}}{W_{dK}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (317)$$

The OEf_d corresponds to the OEf of the steam engine (p. 190).

If it is desired to estimate the probable performance of a turbine, and the OEf is known for similar turbines under similar conditions of operation, the probable water rate per d.h.p.-hour is, from Eqs. (314) to (317),

$$W_d = 2545 \div (\Delta E \times OEf_d), \quad . \quad . \quad . \quad . \quad (318)$$

and per kilowatt hour it is

$$W_{dK} = 3411 \div (\Delta E \times OEf_K). \quad . \quad . \quad . \quad . \quad (319)$$

In very large turbo-generator outfits the value of OEf_K should be 0.65 or more. In general the smaller the turbine the poorer the efficiency, as is shown in a very general way in Fig. 236.

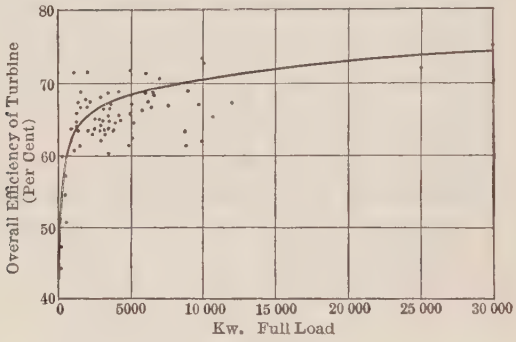


Fig. 236.

(e) The ultimate comparison of the performances of turbines with each other and with steam engines is either on the basis of B.t.u.'s supplied per minute per unit of output, or on the basis of thermal efficiencies. In the *ideal* turbine the **B.t.u. supplied** per h.p. *per minute* are

$$B_I = W_I\,(q_1 + x_1 r_1 + \overline{C}_p D_1 - q_2) \div 60,$$

in which q_2 is the heat remaining in the condensate, which heat is considered as being returnable to the boiler with the feed water (as in Sect. 115 (d)). In the *actual* case the B.t.u. supplied per *d.h.p. per minute* are

$$B_d = B_I/OEf_d = W_d(q_1 + x_1 r_1 + \overline{C}_p D_1 - q_2) \div 60. \quad (320)$$

The B.t.u. supplied per *kilowatt per minute* in the *ideal* case are

$$B_{IK} = W_{IK}\,(q_1 + x_1 r_1 + \overline{C}_p D_1 - q_2) \div 60, \quad . \quad . \quad (321)$$

and in the *actual* case

$$B_{dK} = B_{IK}/OEf_K = W_{dK}(q_1 + x_1 r_1 + \overline{C}_p D_1 - q_2) \div 60. \quad (322)$$

The values of B_{dK} vary from 250 to 800 B.t.u. per minute.

(f) The ratio of the heat delivered as useful energy to that supplied in the steam is the **thermal efficiency.** The thermal efficiency on the d.h.p., as in the case of the steam engine (page 210), is

$$TDEf = \frac{2545}{60 \times B_d} = \frac{2545}{W_d\,(q_1 + x_1 r_1 + \overline{C}_p D_1 - q_2)}, \quad . \quad (323)$$

and based on the kilowatt output it is

$$TDEf_K = \frac{3411}{60 \times B_{dK}} = \frac{3411}{W_{dK}\,(q_1 + x_1 r_1 + \overline{C}_p D_1 - q_2)}. \quad (324)$$

(g) Fig. 237 shows typical curves for a large turbine-generator outfit. It is seen that the curve of *total steam consump-*

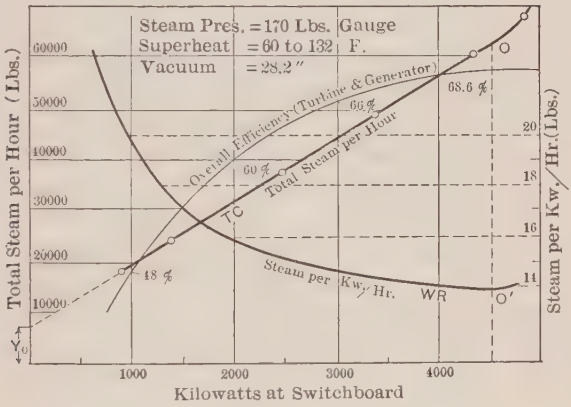


Fig. 237.

tion (T.C. curve) is practically a straight line; and this is a characteristic of such curves for nearly all types of turbines. If, in the figure, the T.C. curve is extended to intersect the Y-axis, the intersept (Y_0) represents the steam required to operate the turbine when delivering no power. It is the amount needed to overcome the friction of the turbine and the "**windage**" (or friction between the turbine disks and the vapor in which they rotate), that required for driving the governor, oil pumps, etc., and that for meeting the losses due to leakage and radiation.

The *water-rate curve* (W.R.), or curve of steam used per kilowatt hour, is also shown in Fig. 237. The water rates at the different loads are obtained by dividing each total consumption by the corresponding kilowatts as found by test, i.e., by dividing

the ordinates of the T.C. curve by the corresponding abscissas. If the T.C. curve passed through the origin, as it would in the ideal case, the W.R. curve would be a horizontal straight line, and the economy of the turbine would be the same at all loads. The greater the Y intercept of the T.C. curve the more curvature does the W.R. curve have, and the greater are the consumptions of steam under light loads as compared with those under heavy loads. It will therefore be noticed that the best economy is obtained when the turbine is operated at its maximum power. As a turbine when operating under its usual load should have some reserve power (or "overload capacity"), it must normally operate at a load and an efficiency less than the maximum. On this account, and because wide fluctuations of load may occur, a flat water-rate curve is desirable.

Many turbines have an auxiliary **"overload valve"** which admits live steam to the low-pressure stages of the turbine when it is considerably overloaded. At such load the T.C. curve and W.R. curve change character, as in Fig. 237 at O and O'.

In Fig. 237 is also shown the *curve of over-all efficiency* of turbine and generator (OEf_K). In this case ΔE has been taken as the available heat in the steam just before it reaches the throttle valve. Thus OEf_K includes the losses entailed by the governor valve throttling the steam, which is the principal reason for the decrease of this efficiency when the turbine output is diminished. Why this loss occurs is explained in (**k**) of this section.

(**h**) Fig. 238 shows on a Mollier chart an expansion line starting with dry saturated steam at p pounds pressure and

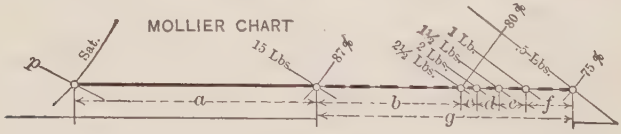


Fig. 238.

extending to various lines of terminal pressure. It is seen that expansion to 15 pounds pressure theoretically makes available heat represented by a, and that further expansion to one-half pound absolute back pressure would add to this an amount of heat represented by g. Thus, if steam from an ideal noncondensing engine or turbine is expanded in a second ideal turbine to one-

half pound absolute pressure, the total power obtainable would evidently be nearly twice (in this instance) that derived from the noncondensing unit. Many **"low-pressure"** or **"exhaust-steam" turbines** are operated with steam received from an engine at about atmospheric pressure, and these in many instances give as much power as do the engines which furnish the steam.

Again referring to the expansion line in Fig. 238, and starting with terminal pressure $2\frac{1}{2}$ lbs., it is seen that the succeeding half-pound drops are accompanied by the heat increments lettered *c*, *d*, *e*, and *f*, the amounts of which, in this particular case, increase the available energy respectively in the percentages 4.7, 5.6, 6.3, and 10. It is apparent that these heat increments rapidly become larger as the back pressure is lowered, hence a one-inch change in vacuum from 28 to 29 ins. is much more effective than one from 26 to 27 ins. The actual case would, of course, differ somewhat from the ideal, but the real gains from improving the vacuum are about proportional to the theoretical; hence, with steam turbines, it is desirable to use as low back pressures as the other considerations will permit.

(i) In the Tϕ-diagram in Fig. 239 the Clausius cycle is superimposed on the Rankine, *ge* being the constant volume line at release in the latter. Now, by decreasing the back pressure, with temperature reduction from T_2 to T_3, the increased amount of heat made available in the ideal turbine is shown by area *abcd*, whereas in the engine it is only *aefd*. Actually there is still greater difference between the gains, for in the engine the increased range of temperature augments the loss due to cylinder condensation, whereas in the turbine there is no equivalent to such condensation since the steam flows continuously in the same direction, and therefore constantly comes in contact with parts which have previously become heated to its own temperature. Evidently, then, the turbine can use very low back pressures to better advantage than the engine, other things being equal.

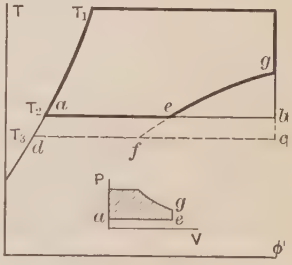


Fig. 239.

(j) The gain due to using **superheated steam** is illustrated on the Mollier chart in Fig. 240. In expanding from dry satu-

rated steam at p pounds pressure to 1 pound pressure, B.t.u. represented by ΔE are seen to become theoretically available, whereas in expanding through the same pressure range but starting with steam superheated 150 degrees, B.t.u. represented by $\Delta E'$ are made available. If p is 165 pounds absolute, a gain of nearly 11 per cent is effected per pound of steam, and only about 0.9 as much steam would be used as with saturated material. The gain in fuel economy is not in this proportion, however, as additional heat was supplied per pound of steam in superheating it. The heat (above 32 degrees) of 1 pound of the superheated steam is seen to be 1277 B.t.u., and in the case of the saturated steam it is 1195. If the feed water is at 72° F., it contains about 40 B.t.u. above 32 degrees. The fuel used per pound in the two cases will then be in the ratio $(1277 - 40) \div (1195 - 40) = 1.07$. Thus the heat supplied by the fuel per unit of work when superheated steam is used is $0.9 \times 1.07 = 0.96$ times that needed with saturated steam, and the theoretical saving in fuel is 4 per cent in this instance. The actual saving may be greater than this, for superheating results in the steam having less moisture after the adiabatic expansion, and the presence of moisture somewhat increases the friction that the steam encounters in passing over nozzle and blade surfaces.

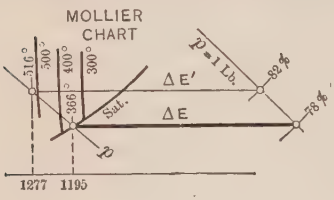


Fig. 240.

In the steam engine, superheating may effect greater improvement in economy than occurs in the turbine, because of its influence in preventing cylinder condensation.

(k) It is quite common practice to decrease the power output of a turbine by throttling the steam supply. This process not only reduces the amount of steam, but lowers its pressure and changes its entropy in such manner that the heat available per pound with any fixed terminal pressure is reduced despite the fact that the total heat of the steam remains the same. This is illustrated in Fig. 241, in which ΔE is the heat available per pound of steam before throttling, and $\Delta E'$ is that after throttling, the total heat ΔQ_1 of the steam being

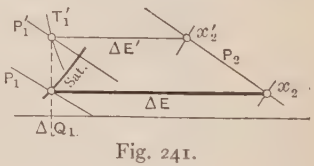


Fig. 241.

initially the same in both cases. It is therefore evident that the throttling process must theoretically decrease the economy of the turbine.

181. Thermodynamics of Actual Turbines. In the energy stream of Fig. 242, ΔE is the heat that would be made available for doing work when there is complete expansion of 1 pound of

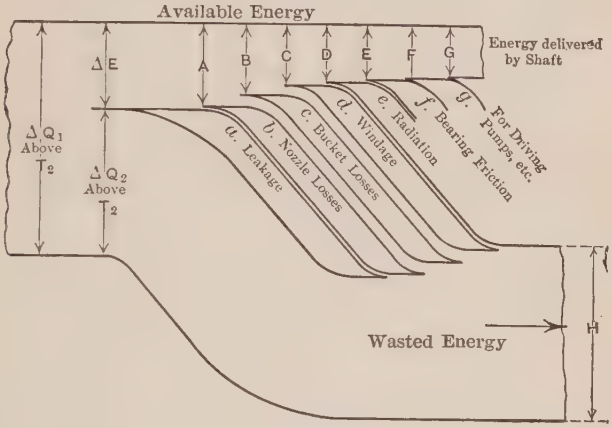


Fig. 242.

steam through the nozzle, or nozzles, of a single stage of an ideal turbine, and ΔQ_2 is the unavailable, or waste, heat.

(**a**) In the actual case, some of the steam may not pass through the nozzle, for there may be **leakage** to the exhaust. For example, in Fig. 234 some of the steam may leak from chamber *2* to chamber *3* through the clearance space *a* between the third diaphragm and the shaft. This leakage loss (which may represent from zero to 5 per cent, or more, of the total energy) is shown by stream line *a* in Fig. 242, and the energy still available for doing work is represented by *A*.

(**b**) Because of the frictional resistance offered by the nozzle walls, and because of eddy currents, etc., all of the heat theoretically made available by the steam expanded through the nozzles is not converted into kinetic energy of the jet. The portion of ΔE not utilized remains in the steam as heat; hence in the figure the **nozzle loss** *b* is shown as subtracted from the available energy and added to that wasted. This loss may be

from 3 to 15 per cent of the total available energy. The energy still available is shown in the figure by B.

(c) Similarly, not all of the kinetic energy of the jet is abstracted by the turbine buckets. The remainder, or **bucket loss,** which may be from 10 to 30 per cent, is reconverted into heat by eddy currents and by the reduction of velocity in the turbine chamber, and this heat is added to that already in the steam before it reaches that point. This loss is represented by c in the figure, and the energy still available, by C.

(d) Further, because of the **"windage,"** or friction between the rotor and the enveloping vapor, not all of the energy absorbed by the buckets is transmitted to the turbine shaft. This loss may be from 2 to 8 per cent with the high velocities of rotation prevailing. This frictional energy is converted into heat by the eddy currents set up in the vapor, and this heat is added to that already stored in the vapor, as shown at d in the figure. The energy still available for doing work is shown by D.

(e) The heat not utilized remains in the steam and is shown by H in the figure. If the steam from this casing is used in another turbine, or as another stage of the same turbine, the diagram of energy flow for this second element would also resemble Fig. 242, but the initial width of the steam line would be H.

(f) In addition to the foregoing, there are the **radiation loss** and the **mechanical losses** from bearing friction and (possibly) from the driving of oil pumps, governor, etc. These are shown at e, f, and g. G represents the energy finally delivered by the shaft. The ratio of G to ΔE is the **over-all efficiency** of the turbine* (not including the generator).

(g) Losses a, b, c, d, and e constitute the equivalent of the cylinder losses in the steam engine; hence, the ratio of the heat shown at E to ΔE may be called the **cylinder efficiency** (IEf).

(h) Fig. 242 will also apply qualitatively to *multistage turbines* considered as a whole, in which case a, b, c, d, and e show the combined losses of all stages.

(i) On the Mollier chart in Fig. 243, let the initial state of the steam be shown by point 1, with pressure p_1, entropy ϕ_1, quality x_1, and associated heat ΔQ_1 per pound. In the ideal case, after expansion through the nozzle to a pressure of p_2 pounds per square inch, the state point would be at 2, with

* This is sometimes called the "shaft efficiency."

entropy ϕ_1, quality x_2, and associated heat ΔQ_2. The heat theoretically made available is shown by ΔE. In the real case, as has been seen, only a part of ΔE is actually delivered to

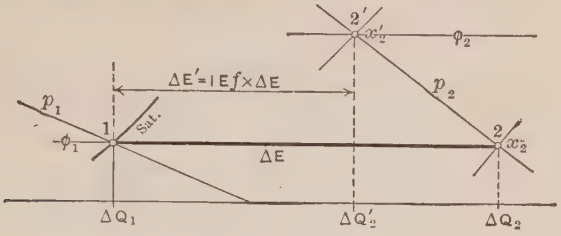


Fig. 243.

the shaft by the wheels or drums. This amount is shown by $\Delta E' = (IEf \times \Delta E)$ in the figure. Evidently the heat remaining in the exhaust steam is shown by $\Delta Q_2' = (\Delta Q_1 - \Delta E')$.

With this amount of heat in the exhaust steam and with the terminal pressure P_2 as before, the state point showing the condition of steam in the actual case must be at $2'$, the point on the pressure line having heat value equal to $\Delta Q_2'$. Thus the actual condition of the exhaust steam is such that the quality is x_2', the entropy is ϕ_2, and the heat above 32 degrees is $\Delta Q_2'$. This is the condition of the steam exhausted to the condenser or to the atmosphere, or to the next stage, as the case may be.

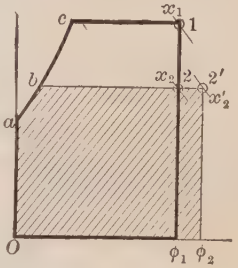


Fig. 244.

(j) Fig. 244 is a Tϕ-diagram corresponding to the Mollier chart in Fig. 243 and is similarly lettered. ΔQ_1 is shown by the area bounded by heavy lines, ΔQ_2 by area $Oab2\phi_1$, and $\Delta Q_2'$ by the hatched area. $\Delta E'$ is the difference between the areas ΔQ_1 and $\Delta Q_2'$ and is not shown directly by any area on the diagram.

182. The Dynamics of Impulse Steam Turbines. (a) In discussing the dynamics of turbines, it is necessary to distinguish between the "absolute" velocity and the "relative" velocity of the jet of steam. **Absolute velocity** is the linear speed (v) of the jet with respect to things that are stationary; the **relative velocity** (R) is the speed of jet relative to the buckets, which themselves are moving with a velocity u.

(b) The available energy of w pounds of steam flowing through the nozzle per second is $w \times 778 \times \Delta E$, and the kinetic energy which it imparts to the jet is

$$KE = \frac{wv^2}{2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (325)$$

in which v is the absolute velocity of the jet in feet per second. Hence, if the nozzle efficiency is Ef_n,

$$Ef_n \times w \times 778 \times \Delta E = \frac{wv^2}{2g},$$

from which the velocity of the jet is found to be (feet per second)

$$v = 223.8 \sqrt{\Delta E \times Ef_n}. \quad . \quad . \quad . \quad . \quad . \quad (326)$$

(c) To completely utilize the kinetic energy of the jet in an impulse turbine, the absolute velocity of the jet must of course be reduced to zero (regardless of the final direction of motion), and it is the function of the blades on the rotor to perform this reduction and receive the energy. If, after passing over the blades, the jet still has velocity (v_2), it is evident that there is loss of energy due to the residual velocity equal in amount to

$$KE_2 = \frac{wv_2^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (327)$$

(d) If in Fig. 245 the jet has an *absolute* velocity v_1 and the bucket has an absolute velocity $u = v_1/2$ in the same direction, the *relative* velocity of jet to bucket is $R = v_1/2$ as it enters. Then if the bucket directs this jet rearwards (opposite and parallel to v_1), the absolute velocity v_2 of the working substance is zero, and the entire energy has been absorbed.

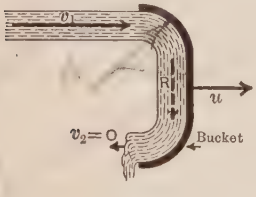


Fig. 245.

Could the friction between the jet and the surface of the bucket, the eddying, and spilling of the working substance, be eliminated, the efficiency of conversion in such a case would be 100 per cent.

(e) If in Figure 246 the line J represents the *absolute* velocity v_1 of the jet and its direction of motion compared with that of the bucket, the direction and velocity of which are shown by u, the *relative* velocity of jet to bucket is shown in amount and direction by R_1, which is found by constructing the triangle

abc with side $bc = u$. If R_2 is the *relative* velocity and direction in which the jet is discharged with respect to the *moving* bucket, then v_2 is the *absolute* velocity and direction of the jet, and its vector is found by constructing the triangle *def* with side $ef = u$. Evidently the presence of this residual velocity v_2 represents a loss of energy which is equal to $wv_2^2/2g$. Hence the bucket efficiency, neglecting other losses, is

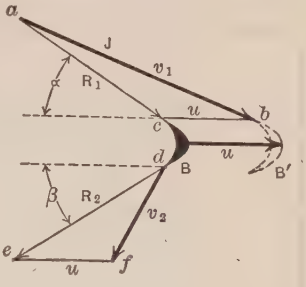


Fig. 246.

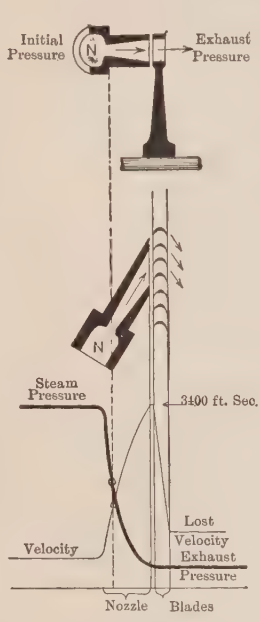
$$Ef = \left(\frac{wv_1^2}{2g} - \frac{wv_2^2}{2g}\right) \div \frac{wv_1^2}{2g} = \frac{v_1^2 - v_2^2}{v_1^2} \quad . \quad . \quad . \quad (328)$$

It will be apparent from Fig. 246 that v_2 can never be made zero if v_1 and v_2 are not both parallel to u, and that unless this is the case the bucket efficiency must be less than unity. It will also be evident that v_2 is a minimum, and the efficiency is maximum, when u is of such value as to cause v_2 to be at right angles to u. This value of v can be determined either graphically or mathematically by methods which need not be considered here. If angle $abc = 20°$, which is about as small an angle as can be used when the nozzle is placed at the side of the buckets, and if R_1 and R_2 form equal and opposite angles with the direction of the bucket's motion, u will be about 47 per cent of v_1.

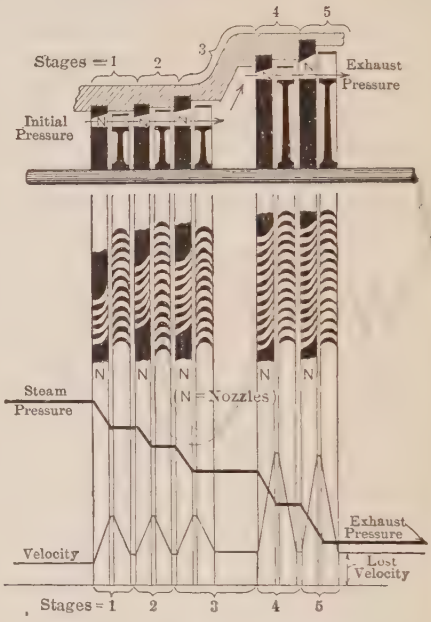
Further discussion of the dynamics of turbines will be given in connection with the descriptions of the various types.

183. De Laval Type of Single-Stage Turbines. — This type of turbine (developed about 1888) is shown diagrammatically at A in the chart given on page 374; and the details of its mechanism are shown in Fig. 247. The velocity diagram resembles Fig. 246, but as the velocity (v_1) of jet issuing from the nozzle may be from 3000 to 4000 feet per second, it is not usually possible to use bucket velocities (u) which correspond to maximum efficiency, for no available materials or possible constructions will withstand such speeds. The bucket velocities are therefore made as high as is safe. The wheels of the 300-horse-power De Laval turbine are about 30 inches in diameter and rotate at about 10,600 r.p.m., with peripheral speed of about 1380 feet per second. The

CHART. — PRINCIPAL COMMERCIAL TYPES OF STEAM TURBINES.



A. De Laval Type.



B. Rateau-Zoelly Type.

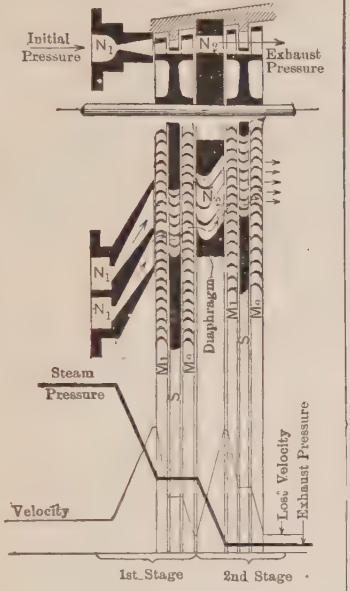
NOTES.

General. In each of the above diagrams the upper portion shows a longitudinal section of the turbine, the middle of the figure represents a transverse section through the buckets and nozzles, and below this are curves which show how the pressure and velocity of the steam vary during the passage of the vapor through the turbine. The pressures and velocities are shown respectively by the ordinates of the heavy and of the light curves.

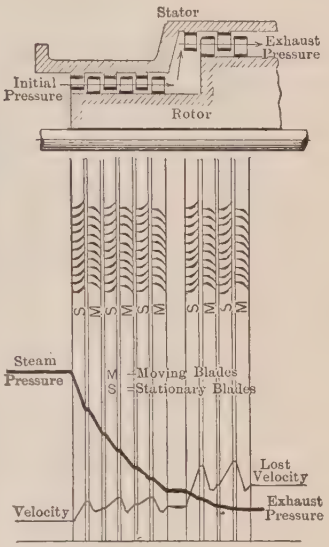
A. De Laval Type (see Sect. 183). In this type of turbine it is to be particularly noted that the full drop in pressure and the entire increase in velocity of the vapor are completed before the jet issues from the end of the nozzle, as shown by the curves; thus there is no expansion of the steam after it reaches the wheel casing. The velocity curve also shows the jet's velocity-decrease resulting from the absorption of the kinetic energy by the buckets, and further shows the residual (lost) velocity associated with the kinetic energy not utilized.

B. Rateau-Zoelly Type (see Sect. 185). Each pressure stage is seen to resemble a single-stage turbine of the De Laval type.

CHART (*Continued*). — PRINCIPAL COMMERCIAL TYPES OF STEAM TURBINES.



C. Curtis Type.



D. Parsons Type.

NOTES (*Continued*).

C. Curtis Type (see Sect. 186). The diagram shows a turbine having two *pressure* stages, each of which has two *velocity* stages. It is seen by the curves that the pressure-drops and velocity-increases occur entirely within the nozzles N_1 and N_2 (i.e., there is no expansion of the vapor in the wheel casings). In each pressure stage the jet first passes over the moving blades M_1 to which it surrenders part of its kinetic energy (thereby losing some of its velocity), and is then guided by the stationary blades S to act on the second set of moving blades M_2, which absorb still more of the energy by further decreasing the velocity of the jet. Thus the (kinetic) velocity energy is absorbed in two steps, or stages, in each pressure stage.

D. Parsons Type (see Sect. 188). Expansion takes place in *both* the stationary and the moving blades, as is shown by the steam-pressure line in the diagram. The steam is accelerated in passing through the first row of stationary buckets; the issuing jets are then retarded by coming in contact with the first moving buckets, to which they surrender part of their kinetic energy; and while passing between these latter buckets the stream is further expanded and issues from them with a reaction. Thus the moving blades receive energy by both impulse and reaction. This process is continued in each of the succeeding pairs of stationary and moving sets of blades.

5-horse-power turbine has a wheel about 4 inches in diameter, the r.p.m. are 30,000, and the rim speed is 515 feet per second.

To provide the maximum theoretical strength, the smaller wheels have sections resembling that in Fig. 247 at (*a*); while the larger wheels are without central hole, the shaft being made in two parts, each fastened to the side of the wheel by flanges. The buckets and the method of attaching them to the wheel are shown at (*b*) in the figure. The flanges on the bucket tips form a continuous **"shroud ring,"** and this prevents the jets from flattening and "spilling" over the ends of the blades.

Although the wheels are balanced with the greatest care, the gravity axis never exactly coincides with the geometrical axis of the shaft. To prevent difficulty which might arise with such high speeds from this lack of balance, the shaft is made slender and flexible so that the wheel can "gyrate" about its gravity axis. Owing to the high speed the "torque" on the shaft is small and a small diameter is therefore permissible.

In most instances the rotative speeds are too great to permit of "direct connection" to the generator, pump, or other machine which is to be driven, hence reducing gears of ratio about 10 : 1 are used.

To obtain continuity of action and noiselessness, the gears are of the opposed "herring-bone" type, with very narrow teeth, which are cut and adjusted with extreme accuracy. The pinion may drive either one or two pairs of large gears, each of the pairs delivering power independently. The power is delivered from the gear shaft through a flexible coupling, the bushings shown black in the figure being made of rubber.

The governor shown at *e* is of the centrifugal fly-ball type. As the weights *W*, *W*, (pivoting on knife-edges at *P*) fly out due to centrifugal force, the rod *R* is moved longitudinally, thus moving the bell crank *L* (in view (*c*)) and regulating the amount of opening of the governor valve *S* (which is vertical on actual turbines). Thus the turbine is throttle-governed. There generally are several nozzles like *d* around the periphery of the wheel, and these are provided with hand-shut-off valves. If the load on the turbine is very small, it is better to close some of these valves, so that the nozzles remaining in action may operate at or near their maximum capacity (the most efficient condition) rather than have all the valves in operation with steam greatly throt-

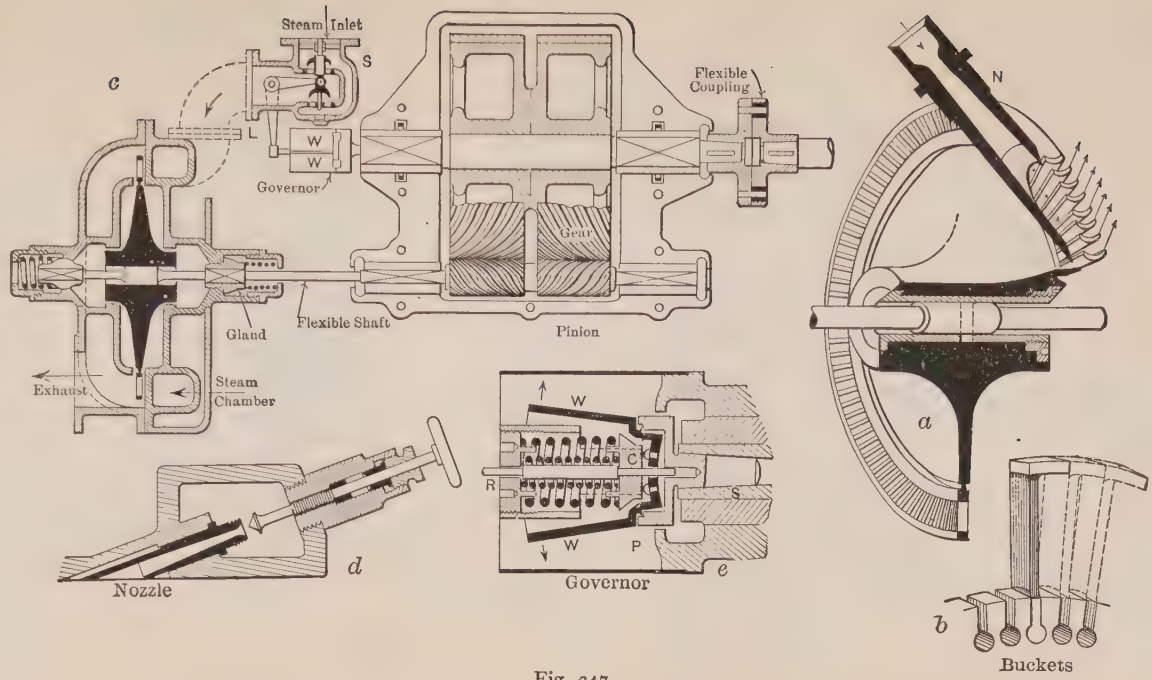

Steam Inlet
S
c
L
W
W
Governor
Flexible Coupling
Gear
Pinion
Gland
Flexible Shaft
Exhaust
Steam Chamber
Nozzle
d
W
C
R
S
W
P
Governor
e
N
a
b
Buckets

Fig. 247.

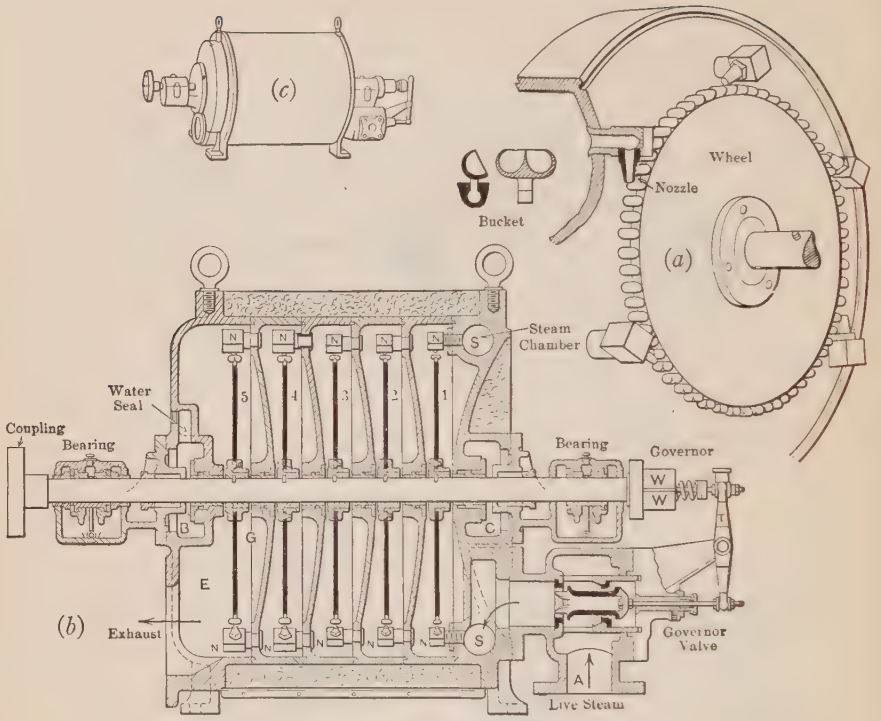


Fig. 248A. Pelton Type.

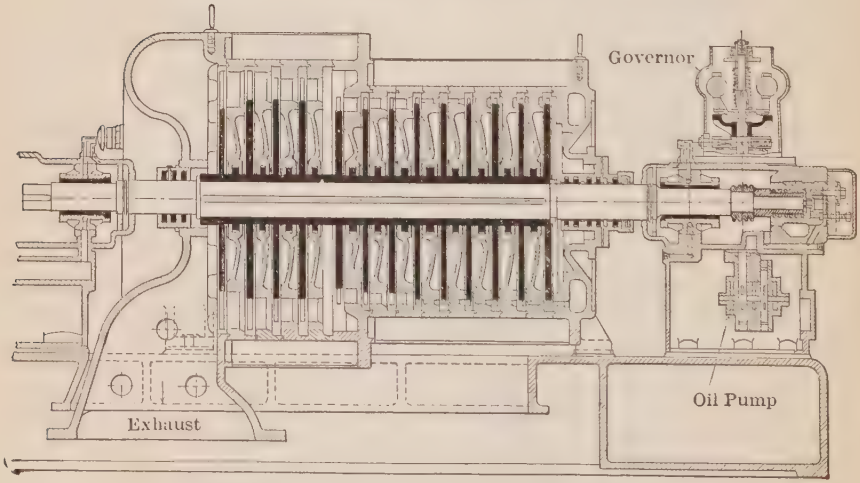


Fig. 248B. Multistage Impulse Turbine (Kerr).

tled (with the accompanying loss). Sometimes there are two sets of nozzles, one to be used when operating condensing, and the other when noncondensing.

184. Pelton Type of Steam Turbine. *Single-stage* impulse steam turbines, with buckets like those used on Pelton water wheels, may be built; but the same difficulties are encountered in them that appear in the De Laval type of single-stage turbine.

By making the turbine *multistage*, and using a sufficient number of stages, these difficulties may be avoided, the jet velocities may be reduced to twice the bucket speeds that can be used safely, — thereby obtaining the highest bucket efficiency (see Section 182 (d)), — and the rotative speeds may be made such as to permit the direct driving of electric generators, centrifugal pumps, blowers, etc., without the use of gearing.

Fig. 248A shows the principal elements of a turbine which is of this type. In this figure (*a*) shows one wheel, the nozzles (one in section), the section of the casing of the adjacent stage of higher pressure, and the bucket. The longitudinal section of the turbine is shown in (*b*). The steam passes from *A* to the chamber *S*, thence through nozzles *N* to the first stage, 1, where the jet impinges on the buckets on the wheel, the section of which is shown black. From the first stage the steam passes in like manner through the nozzles *N* in the diaphragm, to act on the buckets of the second wheel; and so on through the turbine until the steam is exhausted at *E*. To prevent the possibility of any leakage of air through the stuffing boxes at *B* (which would affect the vacuum), a chamber is provided which can be filled with water (forming a "*water seal*") or with steam at pressure slightly above atmospheric. The governor and governor valve are somewhat similar to those of the De Laval turbine. These turbines, formerly known as the "Kerr," have been replaced by the type shown in Fig. 248B.

185. Rateau Type of Steam Turbine. Turbines having from 20 to 40 stages arranged somewhat as in Fig. 234 were developed by Professor Rateau of Paris in 1897. The nozzles, instead of being of circular cross-section, are rectangular, and are grouped closely together so that the intervening walls are thin plates of uniform thickness. The buckets on all wheels, except the last

few, are of the same length. The group of nozzles in the first diaphragm extends over a short arc, that in the next diaphragm is a little longer, and so on; thus as the steam passes through the turbine the circular arc covered by the nozzles and the passage areas increase in size. (See B in chart on page 374.)

The **Zoelly turbine** is similar to the Rateau, except that (1) about half as many stages, and higher nozzle and bucket speeds, are used; (2) in all the diaphragms the nozzle bands extend farther around the peripheries; and (3) the radial widths of the nozzle groups, and the lengths of blades on the wheels, increase from one end of the turbine to the other. Fig. 248B shows a modern turbine of the Rateau-Zoelly type.

186. Curtis Type of Steam Turbine. — Referring to Fig. 246, it is seen that the energy loss from the residual velocity, v_2, is quite large. Curtis (in 1896) patented the arrangement whereby the jet, with this residual energy, is directed to act on other sets of rotating blades, from which it departs with residual velocity much less than in the previous case. This process is termed **"velocity compounding."**

Theoretically, this process may be continued indefinitely, and the final residual velocity may be reduced to any desired value.

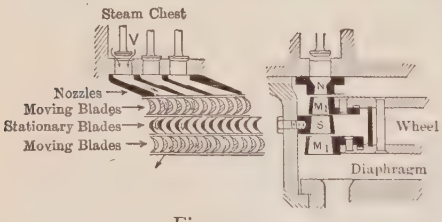


Fig. 249.

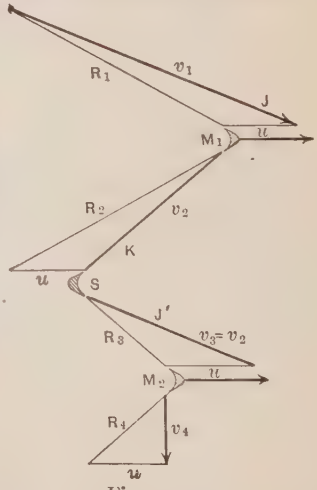


Fig. 250.

In practice, however, the bucket friction and other losses make it inexpedient to use more than two or three rows of rotating blades per stage. Fig. 249 shows the arrangement of a single stage having two rows of moving blades (M), with one set of stationary ones (S) between, all receiving steam from a set of nozzles (N), each nozzle being controlled by a separate valve. Fig. 250 is the corresponding ideal velocity diagram. The velocities R_1, R_2, and v_2 are found in the same manner as in

Fig. 246. The stationary blade S turns the discharge jet K to the direction J' so as to cause it to act on the bucket M_2 with velocity $v_3 = v_2$ (neglecting losses). The velocity diagram $v_3 R_3 R_4 v_4$ is constructed in the same manner as in Fig. 246, and v_4 is the final velocity, the corresponding residual energy (loss) being $wv_4^2/2g$.

As the steam expands fully in passing through the nozzle, the pressure throughout the casing of the stage is uniform. This and the velocity variation are shown in diagram C on p. 375.

The smaller turbines of this type usually have but one stage, while the larger ones have from two to five "*pressure stages*" separated by diaphragms, each diaphragm containing the nozzles for the following stage. These turbines have either horizontal or vertical shafts. In the latter arrangement, which is shown in Figs. 251 and 252, the shaft is

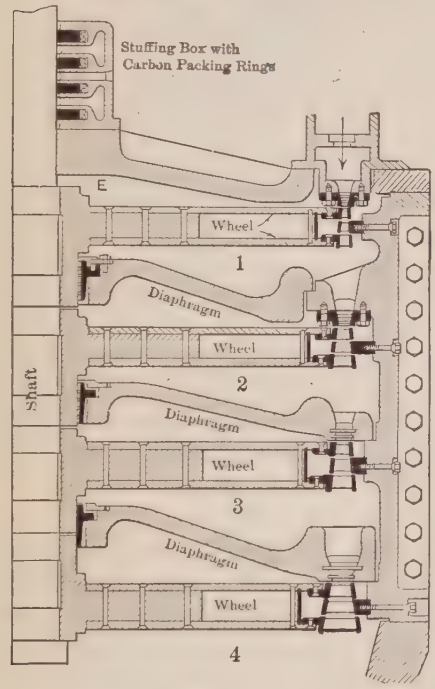


Fig. 251.

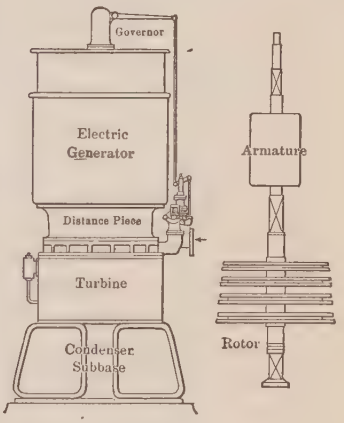


Fig. 252.

supported by a "step bearing," to the center of which oil is supplied at sufficient pressure to support or float the shaft and all parts fastened to it.

Fig. 251 shows diagrammatically a four-stage turbine in which the steam enters at the top and exhausts at the bottom. Such turbines rest on a subbase, which is either connected to the condenser or itself forms the walls of a surface condenser, as in Fig.

252; the generator is placed above the turbine and the governor is mounted on the upper end of the shaft. Fig. 253 shows one arrangement of step bearing, and a portion of the rotating

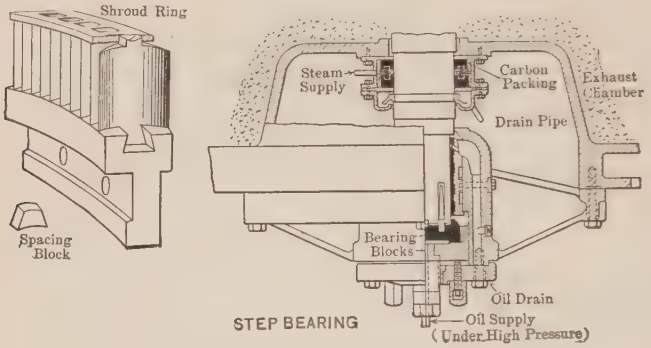


Fig. 253.

"bucket segment," with buckets held in place by "*dovetails*." The buckets are separated by "*spacing blocks*" and their tips are riveted to shroud rings.

On large turbines the governor usually moves a small **"pilot valve"** which controls the position of a hydraulically operated piston, the rod of which moves a shaft having cams which open or close the nozzles of the first stage. Thus the power output of the turbine depends on the number of first-stage nozzles in action. The governing is by the method of **"cutting out nozzles"** or cutting them in.

187. Velocity Compounding with a Single Row of Rotating Buckets. Instead of using a second set of rotating blades in an impulse turbine to abstract some of the energy remaining in the jet when it leaves the first set, as is done in the Curtis type of turbine, this energy can be used (in part) by causing the same jet to impinge repeatedly on a single set of blades.

Fig. 254 shows diagrammatically the elements of the "Electra" turbine (European), which is of this type, and has blades perpendicular to the plane of the wheel disk. The full expansion of the steam occurs entirely within the nozzle N, and the guide passages G merely redirect the steam so as to cause it to impinge properly on the buckets. As the volume of the steam remains constant while passing through the guide passages, the

cross-sectional area of these passages must increase as the velocity (residual) of the steam decreases.

The path of the jet, instead of being serpentine as in Fig. 254, may be helicoidal as shown at *a* in Fig. 255. It may be con-

Fig. 254.

sidered that the lower part of this path is in the semicircular buckets of the turbine wheel shown at *b* and *c* in the figure, and that the upper part is in the stationary guides of similar form. With this construction it is possible to obtain good steam econ-

Fig. 255.

omy with low rotative speeds, even though a single wheel be used. The same scheme is applicable to turbines having two or more pressure stages. The power that is obtainable with any wheel is limited by the number of nozzles and guide "blocks" that can be placed around the periphery.

The forerunner of this type of turbine was the **"Riedler-Stumpf"** turbine (European), with double semicircular buckets like the Pelton.

In Fig. 256 is shown a **Terry** turbine with casing opened. The method of operation is as shown in Fig. 255. Flange *B* couples to the facing *B'* when the turbine is closed, and valve

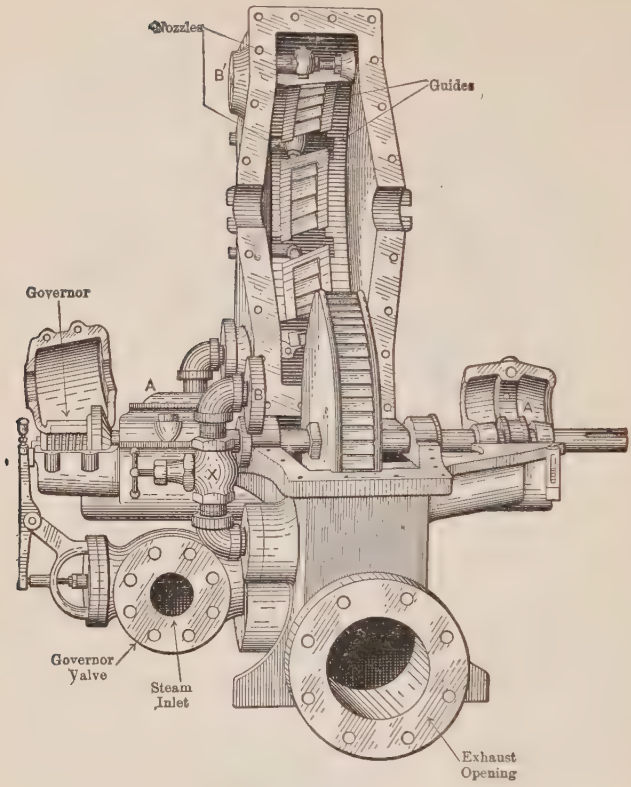


Fig. 256.

X can be used to shut off some of the nozzles when the load is small. The casing is subjected to the exhaust steam only.

The **Sturtevant** turbine, Fig. 257, operates in a similar manner, but is of somewhat different construction. The helical and serpentine paths are used in several other turbines.

188. Reaction Turbines. (a) A simple reaction wheel (similar to Hero's) is shown in Fig. 258. The pioneer developers (De Laval and Parsons) of the modern steam turbine and many other inventors have tried to produce a commercial form of turbine based on this principle, but without success. Experienced designers now recognize the fact that other forms are better for most purposes. The sectioned part *a* in the figure

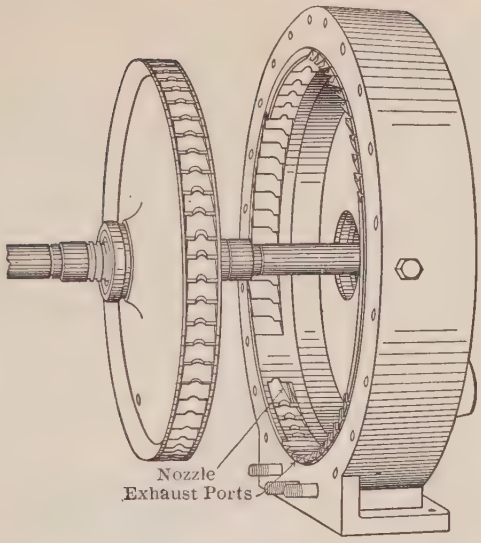


Fig. 257.

constitutes a rotating nozzle of the converging type, corresponding to a small pressure drop from P_1 to P_2.

(b) Another simple reaction turbine is shown in Fig. 259, with blades mounted on the periphery of a disk, or drum, which is arranged to rotate about axis XX. It is seen that the space be-

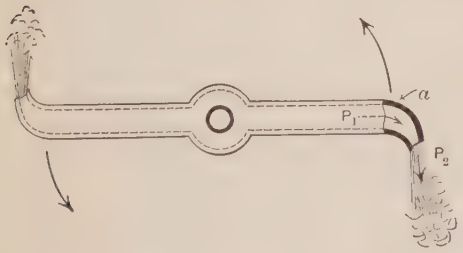


Fig. 258.

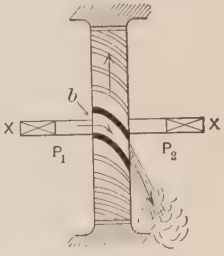


Fig. 259.

tween the blades, as shown at b, has the same form as the nozzle a in Fig. 258; hence there are as many rotating nozzles as there are spaces between blades.

In this arrangement there is a **"full peripheral discharge"** of the steam around the entire circumference, and it is important to note that there is a difference between the pressures P_1 and

P_2 on the two sides of the disk, a condition contrary to that present in the impulse type of turbine.

(c) Fig. 260 may be used to show certain features of the modern type of reaction turbine. Between the tips of the blades, on the drum, and the casing there is necessarily a *radial clearance* space, and because of the inequality between the pressures P_1 and P_2 *leakage* occurs through this space. This clearance is of course always made the minimum practicable. The relative amount of leakage is evidently dependent on the ratio of this annular space to the passage area between blades; thus, the longer the blades are, the less the leakage, with the same

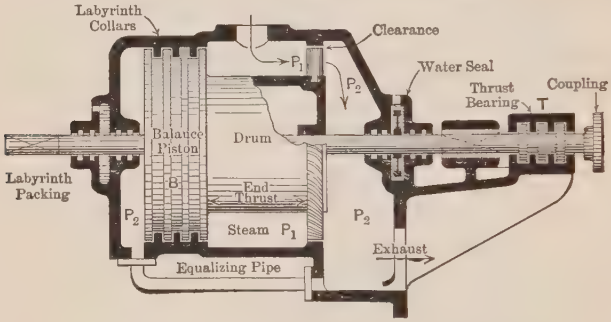


Fig. 260.

clearance. If the peripheral diameter is decreased, not only is the annular space reduced, but the blades must be lengthened to maintain the same passage area between them; hence there is a twofold reduction in the leakage accompanying such change.

(d) It is apparent that the difference between pressures P_1 and P_2 in Fig. 259 causes an end thrust on the shaft. The same is true of the arrangement in Fig. 260. This thrust may be resisted (1) by the thrust bearing T in this figure; or (2) by the balance piston B, which presents to the pressures P_1 and P_2 areas equal to those exposed by the blades and drum end; or (3) by using a "double-flow" arrangement of drum wherein there are two similar rows of blades having discharges which are equal but opposite in direction and hence give opposite end thrusts. In any case there must be a thrust bearing to maintain the rotor in its proper position.

(e) The leakage between the piston B and the shell is usually reduced by employing mating collars, as in Fig. 260, which

form a "labyrinth passage" which becomes more or less sealed by the moisture present in the vapor.

(f) In Fig. 261, R_2 represents the velocity of the jet relative to the rotating blades, u is the blade velocity, and v_2 is the absolute residual velocity of the jet. In practice the velocity u is rather low (usually from 150 to 300 foot-seconds), hence the heat drop per stage is relatively very small.

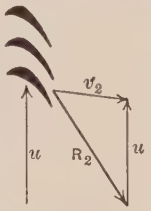


Fig. 261.

(g) The so-called **Parsons' "reaction turbine,"** besides having the rotating reaction blades similar to those in Fig. 260, has stationary guide blades which act as nozzles, the jets from which impinge on the rotating blades. Hence such turbines combine the impulse and the reaction principles.

Fig. 262 shows such an arrangement, S and M being respectively stationary and moving blades. It is seen that not only is there leakage at the tips L_1 of the moving blades, but also at the ends L_2 of the stationary ones. These turbines are made multistage, with stationary and rotating blades alternating. The action of the steam on the moving blades is twofold: (1) The direction of the jet is changed, and if no other action took place it would leave with low velocity—thus there is an impulse action; and (2) the steam expands while passing through the moving blades and acquires velocity by virtue of that expansion, so that when discharged rearwards there is a reaction effect. The residual velocity of the jet leaving the rotating blade is redirected and increased by the next stationary blades and discharged against the next row of moving blades, and so on from one end of the turbine to the other. (See D in the chart on page 375.)

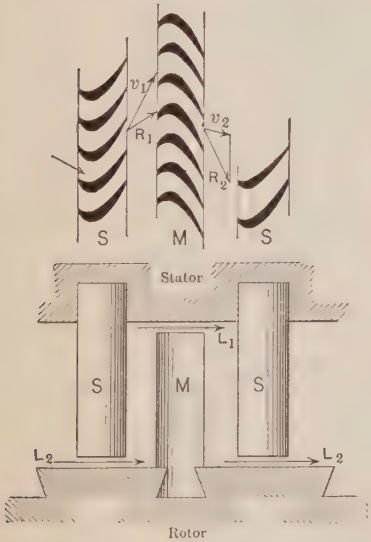


Fig. 262.

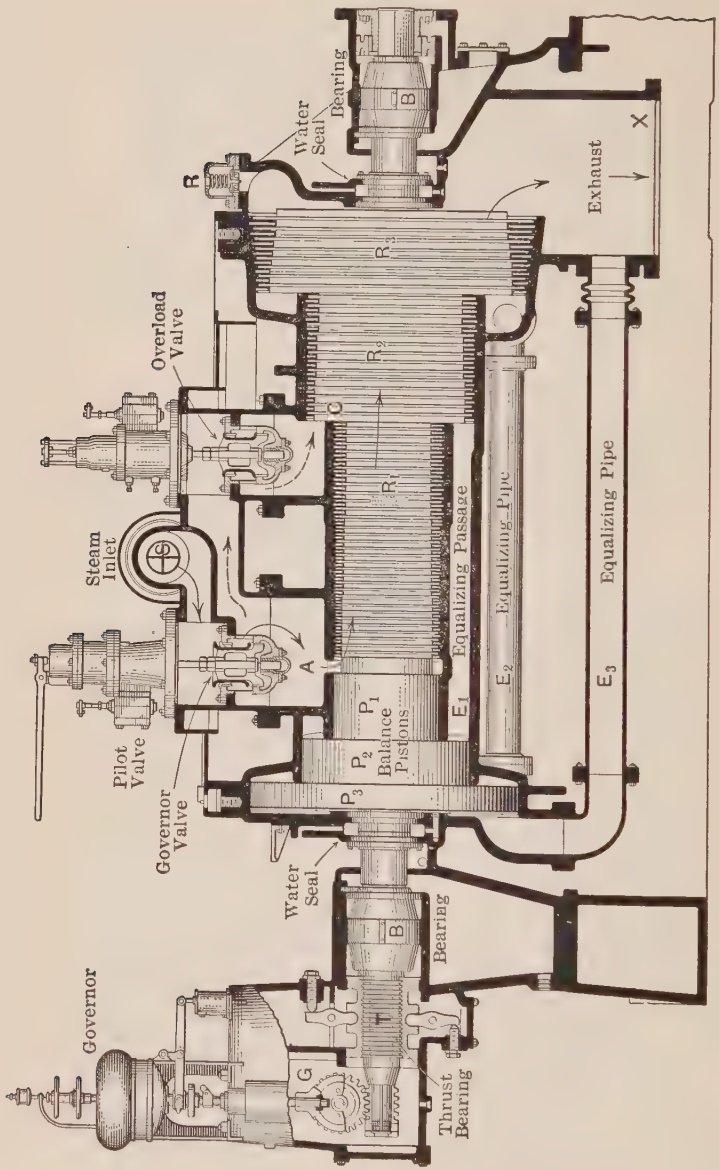

Governor
Thrust Bearing
Bearing
B
G
Water Seal
Governor Valve
Pilot Valve
Steam Inlet
Overload Valve
R
Water Seal
Bearing
B
A
C
P_1
P_2
P_3
Balance Pistons
E_1
E_2
E_3
Equalizing Passage
Equalizing Pipe
Equalizing Pipe
R_1
R_2
R_3
Exhaust
X

Fig. 263.

As the bucket velocities are small, it is necessary to use small heat drops per stage, hence a great many stages are used. As the volume of steam under high pressure is small, the passages between blades must be small, and the blades themselves are consequently short. To reduce the leakage the high-pressure stages are hence made small in diameter compared with low-pressure stages, where the volume of the steam is large and the blades are long. The heat drops in each of the first stages are about 2 or 3 B.t.u., whereas in the last stages drops of about 10 B.t.u. may be used.

(h) One arrangement of the **Westinghouse-Parsons** turbine is shown in section in Fig. 263. After passing the governor valve the steam enters at A and flows between the blades on three cylinders, R_1, R_2, and R_3, which progress in diameter, until it reaches the exhaust opening X. The three *balance pistons*, P_1, P_2, and P_3, with *equalizing pipes* E_1, E_2, and E_3, balance the thrust, and the *thrust bearing* T constrains the rotor to its proper position. The governor moves the pilot valve, which in turn controls the governor valve. The operation is such as to cause the latter valve to constantly move up and down, admitting the steam "by puffs," which vary in duration with the load. If the demand on the turbine becomes more than can be met by all the steam that can flow between the first blades, the turbine's speed will decrease slightly and the governor will then open the overload valve, thus admitting steam to a point (C) where the passage area between blades is greater, so more steam can be used to meet the emergency, although less efficiently than before.

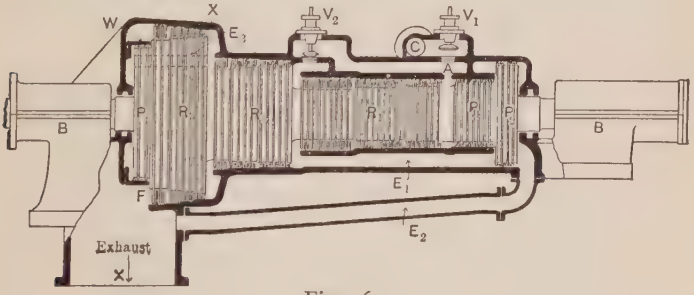


Fig. 264.

(i) Fig. 264 shows the general arrangement of the **Allis-Chalmers-Parsons** turbine, which is of the same general type as

the Westinghouse, but differs somewhat in its arrangement, construction, and method of governing. The largest balance piston P_3 is placed at the exhaust end of the rotor and the governing is by the throttling method.

(j) In multistage turbines the elements in the different stages need not all be of the same type. It is sometimes desirable to use in the first stages that type which operates best with steam at high pressures, and in the remaining stages the type best suited for low-pressure conditions. See Chart on page 394a.

189. Applications of the Steam Turbine. Owing to the high rotative speeds and to the inability to vary these speeds suddenly or to reverse the direction of rotation, there are many fields of "direct driving" which the steam turbine cannot enter.

(a) Driving electric generators, which furnish current for almost an unlimited number of purposes, is the largest field of application.

(b) Turbines are used with direct-driven centrifugal pumps which discharge against low or high heads (circulating pumps, boiler-feed pumps, etc.).

(c) They are used for driving centrifugal air compressors (which are usually multistage), fans, blowers, etc.

(d) In some instances small low-speed turbines have been used for belt driving. Ordinarily it is not feasible to reduce the rotative speeds by use of gearing unless specially designed and constructed for the purpose.

(e) The torque on a turbine shaft is relatively very small, hence turbines are not applicable where a large starting effort is involved.

(f) In many instances steam is available at pressures which are too low to use in an engine, but can be used advantageously in **"low-pressure"** or **"exhaust-steam" turbines,** provided a high vacuum can be maintained in the condenser.

In plants in which engines are operated noncondensing, there can be added turbines of this type to receive exhaust steam at about atmospheric pressure and to exhaust into a condenser having good vacuum. In such cases it is desirable to maintain a pressure slightly above atmospheric in the pipes between the engine and the turbine to avoid leakage of air into the steam

with resultant decrease of vacuum. When the steam is received at about atmospheric pressure, the exhaust turbine will develop a horse power with about 30 pounds of steam per hour, provided the vacuum is good. The probable water rate in any case can be computed by using Eq. (318). With this and the available amount of steam known, the power that can be developed by the exhaust turbine can be readily computed.

Condensing engines can be operated noncondensing with late cut-off, thus giving about their normal power, and the exhaust steam can be used in a low-pressure turbine, the combined outfit thus giving power greatly in excess of (sometimes double) the normal power of the engine alone. Such arrangements usually are more economical than either the engine under the original conditions or a turbine which receives steam direct from the boiler. The less economical the engine, the more heat remains in its exhaust steam for use in the turbine.

(g) If the supply of steam furnished a low-pressure turbine is intermittent, as in a rolling mill, a **regenerator** or **accumulator** can be used to make up any temporary deficiency which may occur. This device consists of a closed vessel which contains water over or through which the steam passes on its way to the turbine. Thus this water becomes heated to the temperature of the steam. Should the supply of steam cease, the steam pressure would decrease and as a result some of the water would vaporize and supply the turbine with working substance at continually decreasing pressure for a short interval of time. Usually there is also provision for supplying steam direct from the boilers, through a reducing valve, in case the normal supply fails for a considerable length of time; and sometimes the turbine has a high-pressure stage which is normally inoperative, but which is brought into action in such an emergency.

(h) **Marine propulsion** is another large field for the application of the steam turbine. A saving in the weight of the turbine and of space occupied can be effected by using high rotative speeds, and the economy can be improved by using high bucket velocities. On the other hand, the propeller on a slow-moving vessel is inefficient if operated at rotative speeds which are high. Hence in applying the turbine to the direct driving of a propeller, a compromise must be effected; evidently the best results should be obtained on high-speed vessels, and such is the case.

There have been invented numerous speed-reducing devices (mechanical, hydraulic, and electric) to be placed between the turbine and propeller shaft, but there is still doubt as to their feasibility, and until such devices are used the application of the turbine will probably be limited principally to vessels of high speed.

Special provision must be made for backing; usually a small "backing element" is placed at the end of the turbine. As turbines are very uneconomical when operating below their normal speeds, they should not be used for low-speed cruising. Sometimes smaller "cruising turbines" are added for such service. Turbines are not satisfactory when much maneuvering must be done, and in some instances a combination of engines and turbines has been used for such service.

190. Advantages and Disadvantages of the Steam Turbine. (a) When operating under *normal load* (i.e., with the usual allowance for overloading), a comparison of the *best performances* does not show that the turbine has any advantage over the engine even when unusually good vacuums are used with the former. It is probable, however, that the average large *condensing* turbine with high vacuum gives better performance than the average large condensing engine. In general, non-condensing and small turbines do not compare so favorably. In many cases it is found that from the standpoint of fuel economy there is little choice between the turbine and the engine, in which cases the selection must be based on other considerations.

Comparison should be made at *normal* load and should be either on the basis of B.t.u. actually supplied per d.h.p.-minute, or of the thermal efficiencies, rather than on the basis of the steam used, unless the conditions of operation are identical.

(b) The *water-rate curve* for the turbine is usually flatter than that of the engine, and hence with widely fluctuating load the average economy is nearer the best for that machine; especially is this the case if the unit is overloaded.

(c) The *space* occupied by the turbine is much less than by the ordinary engine, especially if the latter is horizontal. In some cases, however, this is partly offset by the greater space that may be occupied by the larger size of auxiliary apparatus frequently used with turbines.

(d) The turbines use no *oil internally*, hence the condensate is suitable for direct return to the boilers, and the heat-transmitting surfaces of the boilers, condensers, feed-water heaters, etc., being free from oily coating, operate under best conditions.

(e) Turbines have the advantage of greater *uniformity of rotation*, and can give close speed regulation. If properly "balanced," they are practically free from vibration, hence do not require massive foundations or flywheels.

(f) Other considerations are the first *cost* of turbine and generator (which is generally less than that of the engine-generator outfit), and the cost of auxiliary apparatus (which is often greater with high vacuums). The cost of ground occupied, the building and foundations, the reliability, the cost of condensing water, supplies, attendance and repairs, the allowance for depreciation, etc., must also be considered. Such matters, however, relate to the Economics of Power-Plant Engineering, which will not be discussed here.

(g) There are many fields in which it is necessary to operate at low *angular velocity*, at variable speed, with reversal of motion, or with large starting torque, which the steam turbine cannot enter. There are other fields in which high angular velocity is desirable, or not disadvantageous, and in many of these the turbine is as satisfactory as, or more so than, the engine.

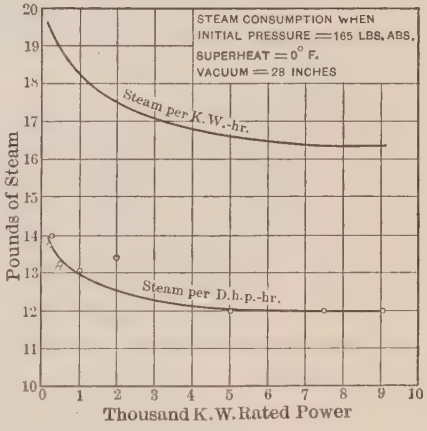


Fig. 265.

191. Steam-Turbine Performance. (a) Fig. 237 shows the general character of the steam-consumption curves for steam turbines, the curve for total consumption being substantially straight. The water-rate curve is usually flatter than that of the steam engine, which indicates less variation in economy with fluctuations of load. Inspection of this curve shows that the best economy is obtained at the maximum load (not at the normal),—

unless there is an "overload valve" which reduces the efficiency when open.

(b) The curves in Fig. 265 show the economies of the larger sizes of standard American turbines when operating under rated loads which are used as abscissas.

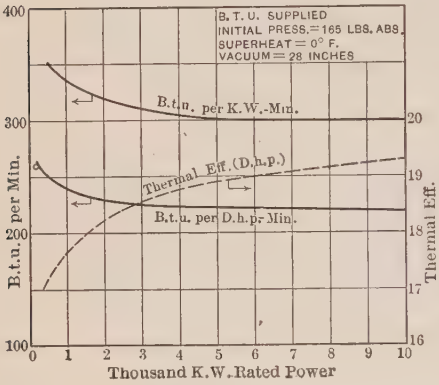


Fig. 266.

These curves are based on Prof. J. A. Moyer's tables,* in which the steam consumptions as obtained by test are reduced to the equivalent values which the same turbines would probably give if operated under pressure of 165 pounds absolute, zero superheat, and 28-inch vacuum. The corrections were made by using the following reduction factors:

For *all types* of turbine:

Per pound of initial pressure, $\frac{1}{10}$ per cent.

Generator efficiency, 91 per cent (300 to 400 kilowatts), 95 per cent (500 kilowatts), 96.5 per cent (1000 to 3000 kilowatts), and 98 per cent (5000 to 10,000 kilowatts).

For *Parsons type:*

Per degree of superheat, $\frac{1}{10}$ per cent (300 to 1000 kilowatts) and $\frac{1}{8}$ per cent (1200 to 7500 kilowatts).

Per inch of vacuum, 4 per cent (300 to 1000 kilowatts) and 3 per cent (1200 to 7500 kilowatts).

For *Curtis type* (may also be used for Rateau and Zoelly turbines):

Per degree of superheat, $\frac{1}{8}$ per cent.

Per inch of vacuum, 7 per cent (26 to 28 inches) and 9 per cent (28 to 29.5 inches).

These correction factors should be used only when the changes involved are slight, otherwise the results may not be reliable.

(c) In Fig. 265, the curve for "steam per d.h.p.-hr." is based on the brake horse power, or power delivered by the turbine

* Page 287, Moyer's "The Steam Turbine." Wiley & Sons.

CHART

Turbines having Combinations of Stages of Different Types

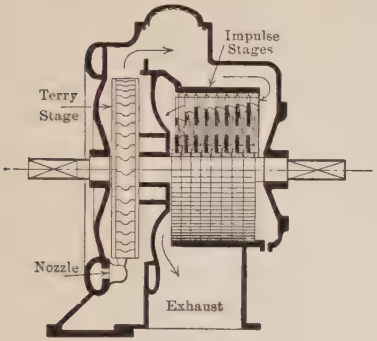


Fig. A. "Return Flow" (Terry).

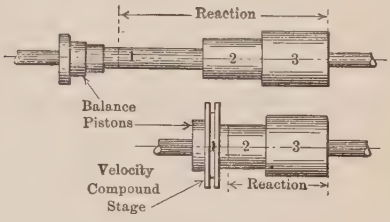


Fig. B. Comparison of Lengths of Rotor (Reaction vs. Combination of Velocity Stage and Reaction).

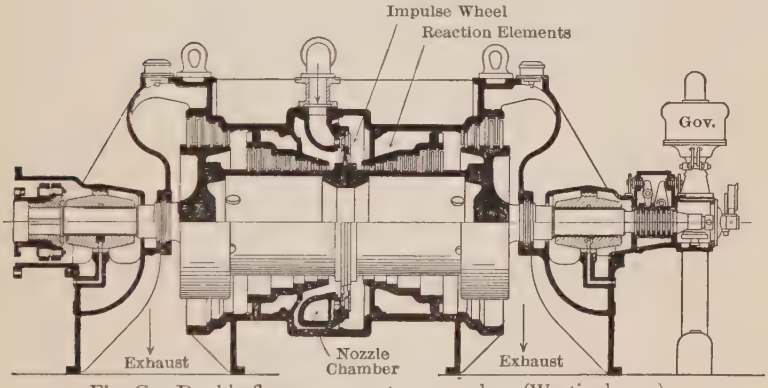


Fig. C. Double flow — 10,000 to 20,000 kw. (Westinghouse).

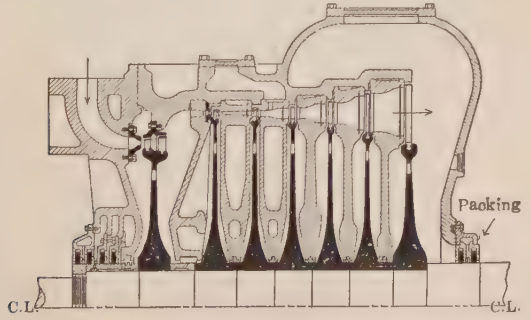


Fig. D. Seven-stage Impulse Turbine (General Electric).

TABLE X. SUMMARY OF PERFORMANCES OF LARGE STEAM TURBINES

(From a "Lefax" data sheet)

1 Type.	2	3 K.W.	4 R.P.M.	5 Initial Pressure, Lbs. Absolute.	6 Superheat. Degrees Fahr.	7 Vacuum (on 29.92" barom.), Inches.	8 Lbs. of Steam per K.W.-Hr.	9 B.t.u. per K.W.-Hr.*	10 Thermal Efficiency. T. Ef.	11 Efficiency Ratio or Overall Ef. O. Ef.	12 Remarks: (Data from Tr. A. S. M. E., 1912, p. 474, except as noted.)
Curtis-Parsons	Range†	1250–11466	750–3000	128–200	60–215	26.41–29	12.56–15.18	15570–18060	21.9–18.9	62.1–71.8	15 Turbines
	Best T. Ef.‡	6000	960	185	198	28.18	12.56	15570	21.9	71.3	
	Best O. Ef.‡	2128	1500	156	120	27.89	13.82	16460	20.7	71.8	
Parsons	Range	3000–6257	1200 1800	156–214	121–259	27–29	11.95–15.4	14980–18433	22.7–18.5	63.5–68.8	6 Turbines
	Best	6257	1210	204	176	29.02	11.95	14980	22.77	68.8	
Curtis-Rateau or Curtis-Zoelly	Range	1545–6565	1220–1500	140–210	89–285	26–29.28	11.43–16.24	14640–19020	23.3–17.9	62.8–68.7	11 Turbines
	Best	6518	1220	198.7	220	29.28	11.43	14640	23.3	68.7	
Zoelly or Rateau	Range	1235–6383	1000–3000	133–221	79–252	26.38–29.25	11.44–16.13	14970–19350	22.8–17.6	62.2–67.5	12 Turbines
	Best T. Ef.	3166	1500	214	276	29.25	11.44	14970	22.8	66.1	
	Best O. Ef.	6383	1000	203	137	27.33	14.3	17150	19.9	67.5	
Curtis	Range	1221–10816	750–3000	127–210	7–276	26.7–29.4	11.77–17.75	15450–20690	22–16.5	61–66.5	11 Turbines
	Range	2000–12460	720–3000	165–207	0–191	27.5–28.65	13.63–16.05	16720–18130	20.4–18.8	64.3–68.2	7 Turbines
	Best T. Ef.	2236	1500	191.6	276	29 34	11 77	15450	22.08	63.6	Tr. A. S. M. E.
	Best O. Ef.	12460	720	207.5	191	27.52	13.63	16760	20.3	68.2	1912, p. 499.
Parsons	Guarantee	25000	750	215	200	28 92	11.25	14200	24.00	72 4	Power Nov. 4, '13. (Commonwealth.)
Parsons Cross Comp.	Estimate	30000	1500 & 750	215	120	29	11.27	13774	24.77	75.76	W. M. Co. 12-31-13 (Interborough).

Columns 8, 9, 10 and 11 do not include losses due to auxiliaries, but do include generator losses.

* Above heat of liquid at exhaust pressure. † Total range of data. ‡ Complete data for the turbine having the best efficiency.

shaft, and does not include the generator losses. The other curve includes the losses of both the turbine and the generator.

(d) Fig. 266 shows the B.t.u. consumptions and the thermal efficiencies corresponding to the water-rate curves of Fig. 265. Values better than here shown can be obtained by using higher superheats, higher pressures, and lower vacuums — especially by using the latter.

Some of the best results so far obtained with *large* turbines are given in Table X.*

(e) *Small turbines* are generally much less economical in the consumption of steam than large ones. In most instances this is largely due to the use of bucket velocities which are much less than those corresponding to the best performance, and which result from using small wheel diameters and low rotative speeds which permit of direct connection to generators, pumps, etc. The economy of such turbines is greatly influenced by a change in bucket speed.

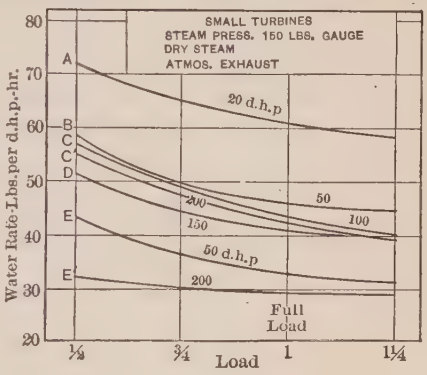


Fig. 267.

Fig. 267 shows curves of steam consumption per brake horse-power for several types of small turbines.† It shows that in general the smaller sizes have poorer economy.

(f) The results of some tests of the 59th Street Power Plant of the Interborough Rapid Transit Co., New York City,‡ can be used to compare the performances of the same power plant operating first without and then with exhaust steam turbines. This plant was originally equipped with condensing engine-generator units each developing 5000 kw. at normal (economical) load, and having a maximum rating of 7500 kw. Subsequently, exhaust steam turbines of 7500 kw. maximum capacity were

* Also see Christie, "Present State of Development of Large Steam Turbines," Trans. A. S. M. E., 1912.

† See Orrok, "Small Steam Turbines," Trans. A. S. M. E., 1909.

‡ Trans. A. S. M. E., 1910, page 78.

added, and these were arranged to operate with high vacuums on steam received at about atmospheric pressure from the engines, which latter were changed to have later cut-offs than before. The addition of the turbine to one of the engine units gave the following results: —

(1) The maximum capacity was doubled.

(2) The load giving the best steam performance (or load under which the unit should normally operate) was increased to about $2\frac{1}{2}$ times its former value.

(3) The average steam economy (between 7000 and 15,000 kw.) was 25 per cent better than that of the old engine unit.

(4) It was estimated that the average steam economy was 13 per cent better than would have resulted by using a high-pressure turbine of the best design in place of the combined unit.

(5) The average thermal efficiency of the combined unit was 20.6 per cent (between 6500 and 15,500 kw.).

CHAPTER XXIII.

EXTERNAL-COMBUSTION GAS ENGINES.

192. Definition. (a) The name **gas engine** is applied to prime movers in which the working substance is material in gaseous form comparatively far removed from the conditions of liquefaction. Hence the working substances may be assumed to obey the laws of ideal gases, and each gas engine follows approximately one of the gas cycles discussed in Chapter VIII.

(b) At present *gaseous working substances* consist of air mixed with other materials such as carbon monoxide, hydrocarbon vapors and gases, water vapor, and carbon dioxide.

(c) The "*hot body*" is approximated in real engines by the burning of fuel in some chamber at a rate which maintains the required high temperature. In some types of gas engines the fuel, and resulting hot "products of combustion," are separated from the working substance by metallic walls through which the heat received by the working substance must pass. Such engines are known as **external-combustion gas engines.**

In other gas engines the fuel and the air for its combustion are burned inside the cylinder of the engine, and the hot products of combustion form the expanding working substance. Such engines are called **internal-combustion gas engines**, or simply internal-combustion engines.

193. The Hot-Air Engine. (a) Many attempts have been made to utilize air as a working medium in external-combustion engines, but only two such engines survive to-day in this country, and they are built only in small sizes and for special service. Inventors have long tried to produce an actual gas engine to work with close approximation to the Carnot cycle. Even if this were possible, it would be unwise commercially because of the excessive cylinder volume necessary for a given output of power.

This may be made clear by drawing, as in Fig. 268, a Carnot cycle, an Ericsson cycle, and a Stirling cycle for comparable

conditions so as to show the greatest volume occupied by the gas in each case. This is best done by imagining one pound of gas (air in the figure) to receive the same amount of heat and to work between the same temperature limits in each case. The efficiency and work done will then be the same in each case, and will be

$$Ef = (T_1 - T_2)/T_1,$$

and

$$\Delta E = \Delta Q_1 \times Ef = \Delta Q_1 (T_1 - T_2)/T_1.$$

The figure shows that the *maximum volume* occupied by the working substance, and hence the necessary cylinder volume, is much greater in the case of the Carnot engine than for either of the others.

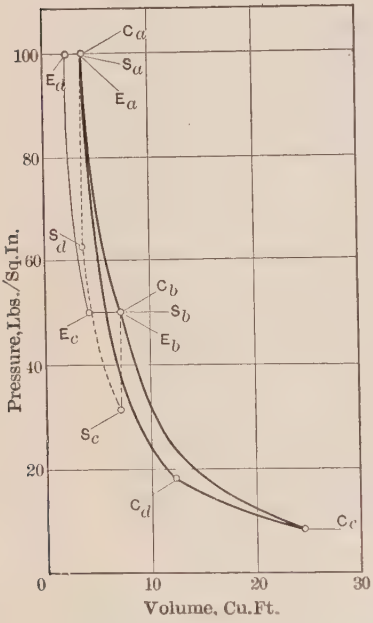


Fig. 268.

(**b**) The cylinder volume, however, determines to a considerable extent the *size* and *cost* of the engine. As will be brought out later, the real external-combustion gas engines operating on the Stirling and Ericsson cycles are almost prohibitively large, and it is therefore obvious that the Carnot gas engine with external combustion must be commercially impossible.

(**c**) The two external-combustion gas engines now in use in this country are the **Rider hot-air engine** and the **Ericsson hot-air engine.** The former approximates the Stirling cycle and the latter the Ericsson cycle. Both of these engines are direct-connected to small water pumps which utilize the net work of the engine. They are simple and satisfactory pumping engines, particularly for farm and suburban use.

(**d**) The *capacity*, that is, the power made available, is *limited* by the *slow rate of heat transfer* between the metallic walls and the more or less quiescent gas; by the *slow rate of heat conduction* in the gas itself; by the *low specific heat* and *density of the gas;*

and by the comparatively *low maximum temperature* at which it is advisable to maintain metal.

In order that gas may absorb or give up heat rapidly, it must pass over the metallic surfaces in a thin stream and at high velocity. This necessitates a large engine. The time necessary for heat exchanges is so great that the engine must run at very slow speeds with few cycles per minute. Obviously, the smaller the number of cycles the greater must be the energy made available per cycle, and hence the greater the size of the engine to deliver a given amount of power.

The effect of low specific heat is to increase the weight of gas necessary for a given heat change, and the low density results in a large volume for a given weight. Both effects increase the size and cost of engine for a given power.

194. Rider Hot-Air Engine. (a) This engine, which approximates the Stirling cycle (Section 54), is shown in Fig. 269, and by comparing it with Fig. 23, reproduced at (*b*) in the upper right-hand corner of Fig. 269, it is seen to have all the parts of the ideal engine following this cycle. In the two figures similar parts are designated by the same letter. The ideal hot body is replaced by a furnace, the gases of which at temperature T_1 jacket the hot cylinder *Y*. The regenerator *R* is approximated by a passage filled with closely spaced plates *HH*. The cold cylinder Y_1 is jacketed by water *X* from pump *P* and is maintained at a practically constant temperature T_2. The water replaces the cold body. By means of connecting rods *J* and *J'* the pistons *D* and *C* are connected to cranks *I* and *I'* which are fastened to the shaft, with the crank for the hot cylinder

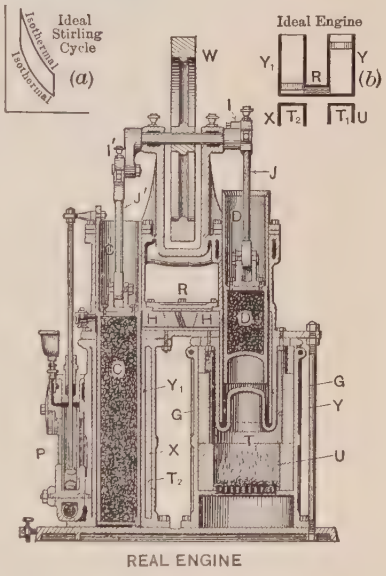


Fig. 269.

leading the other* by about 90 degrees. The crank shaft carries a flywheel W which maintains uniformity of rotation, even though the power, developed and delivered, varies widely.

(b) In the ideal cycle, Fig. 22, and (1) in Fig. 269, it is assumed that during the isothermal reception of heat by the gas, the left piston remains stationary at the bottom of its stroke, while the right piston rises; that during the isothermal rejection of heat the reverse action occurs; and that during the isovolumic changes the two pistons move at such rates as to keep the total inclosed volume constant while the gas passes through the regenerator.

In the actual case these actions are roughly approximated by connecting the pistons to cranks which are nearly at right angles. When either piston is at or near its lowest position, most of the working substance is in the other cylinder, where the piston is at about half-stroke (since the cranks are at about right angles). The material meanwhile is undergoing an isothermal change, which is expansion if the gas is in cylinder Y, or compression if in Y_1. In two intermediate positions of the cranks the pistons are moving with equal and opposite velocities, while the gas is undergoing isovolumic transfer from one cylinder to the other. Thus in the Rider engine the Stirling cycle is roughly approximated, with considerable blending between the various processes.

(c) The *actual diagram* obtained from this engine cannot readily be directly compared with the theoretical, and a reproduction is therefore omitted. The maximum and minimum temperatures are respectively lower and higher than the theoretical, and the corners of the diagram are very much rounded.

(d) If the furnace temperature be assumed at 1500° F., a low value, and the jacket temperature at 60°, a rather high value, the Stirling cycle efficiency is

$$CEf = \frac{T_1 - T_2}{T_1} = \frac{(1500 + 460) - (60 + 460)}{1500 + 460} = 73.5 \text{ per cent.}$$

The actual *thermal efficiency* on the i.h.p. ($TIEf$) is seldom as much as 2 per cent, so that the best *indicated efficiency* is about

$$IEf = 2/73.5 = 0.027 = 2.7 \text{ per cent.}$$

* That is, preceding it in the direction of rotation.

The corresponding efficiency for internal combustion gas-engines and steam engines is generally 50 per cent or more. The *poor economy of this engine* is thus very striking.

195. Ericsson Hot-Air Engine. (a) This engine, which approximates roughly the Ericsson cycle (Section 55), is shown in Fig. 270, in which the parts are lettered to correspond with the

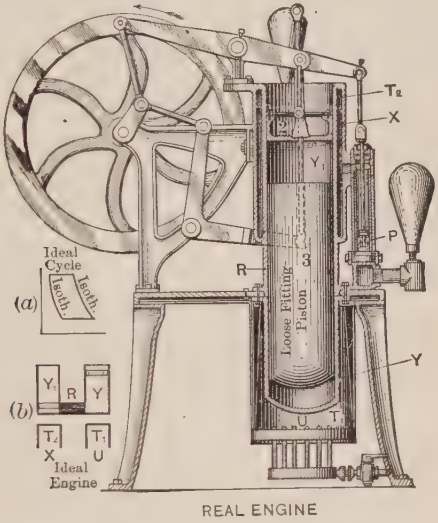


Fig. 270.

theoretical engine shown in Fig. 23. The furnace U replaces the hot body; below the "displacer" piston *3* is the hot cylinder Y, jacketed by the furnace gases, and above it is the cold cylinder Y_1 with jacket X supplied with water from pump P. The upper or working piston *2* transmits the power. When the displacer piston *3* is up, most of the gas is below it in the hot cylinder. Upon descending, this piston transfers the gas to the cold cylinder Y_1 above, and when ascending, returns it to the hot cylinder Y. The function of the regenerator R is performed by the walls of the cylinder and of the loose-fitting displacer piston, between which the gas passes in transferring from one cylinder to the other.

The engine has an ingeniously arranged mechanism which gives such kinematic motion to the displacer and working pistons

as to produce approximately the PV-changes of the theoretical Ericsson cycle.

(b) The conditions of heat transfer are even poorer in this engine than they are in the Rider, and as a result the power developed for the same heat supply, and for same size of cylinder, is only about one-third of that obtained with the other engine.

(c) It is worthy of note that a very large hot-air engine of a different type, which was constructed by Ericsson, gave a thermal efficiency of about 10 per cent. It was, however, enormously bulky and mechanically unsatisfactory.

CHAPTER XXIV.

INTERNAL-COMBUSTION ENGINES.

Methods of Operation.

196. Advantages and Types. (a) Although *external*-combustion engines with gaseous working substance are not generally commercially successful, the *internal*-combustion engine, on the other hand, is widely used, and is capable of giving the highest economies now attained by any type of heat engine.

The success of the internal-combustion engine is chiefly due to the fact that, since the *products of combustion constitute the working substance*, the maximum temperature is that due to combustion; whereas in the external-combustion engine the maximum temperature of the working substance is limited by the capacity of metallic walls to withstand high temperature, and to transmit heat.

In internal-combustion engines with proper design the highest temperature attainable may be used without danger to metallic walls, and it is thus possible to approach theoretical efficiencies corresponding to temperatures of from 2500° to 3000° F. Because of the high pressures that accompany high temperatures, the engines are also small for a given capacity.

(b) During the past twenty years the use of internal-combustion engines has rapidly increased, until now many large power plants depend entirely upon them for power. These engines operate on either the *Otto* cycle or the *Diesel* cycle. Engines following the latter cycle were until recently a more or less special type adapted only to certain limited conditions, but this limitation is rapidly disappearing.

(c) There are two distinct types of engine following the Otto cycle; one requires two piston strokes, and the other four, to complete a cycle. They are known as **two-stroke cycle** and **four-stroke cycle** engines, or improperly as "two-cycle" and "four-cycle" engines. The four-stroke cycle is in more

common use, though it has several theoretical and practical disadvantages as compared with the other type.

197. Cylinder Operations of Four-Stroke Otto Cycle. (a) The heat is evolved *within* the cylinder by the burning of a mixture of fuel gas, or vapor, with air, which supplies oxygen for combustion. The gaseous products of combustion form the working substance, which, after expansion, must be expelled from the cylinder to give place to a fresh combustible charge for the next cycle. The engine is shown diagrammatically in Fig. 271.

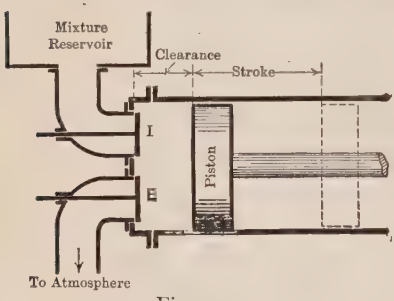


Fig. 271.

(b) Imagine a cylinder as shown in the figure, with an inlet valve *I* and an exhaust valve *E* located in the head and arranged to open inwardly; and assume that the piston is in its extreme left position, that its motion can be controlled as desired, that a cycle has just been completed, and that the "*clearance space*" or "*combustion space*" between the face of the piston and the cylinder head is filled with burnt gases at atmospheric pressure.

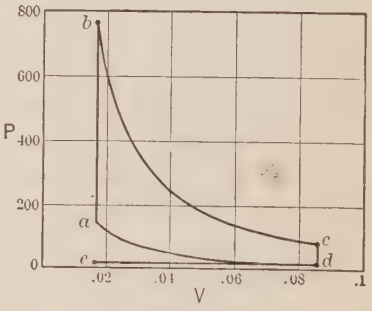


Fig. 272.

Now with the valve *E* closed, and with *I* open to a supply of combustible mixture at atmospheric pressure, the *first stroke* of the piston (to the right) will cause some of this mixture to pass into the cylinder, where it will mix with the burnt gases, and thus diluted will fill the available space at approximately atmospheric pressure. The line *cd* in Fig. 272, at a height of 14.7 pounds per square inch above the horizontal axis, represents this process.

Now imagine the inlet valve *I* closed and the piston moved to the left performing the *second stroke*. During this stroke the mixture will be compressed until, finally, its volume is reduced to that of the clearance space. This compression may be

assumed to be adiabatic, although this would not be absolutely true in any real case on account of the thermal properties of the metallic walls. The ideal process is represented by the adiabatic compression line *da*, corresponding to the similar line in Fig. 26, page 94.

At this point *a* the charge is ignited by an electric spark, or other means, and it may be assumed to burn completely with the piston stationary. This would cause an increase in temperature and pressure corresponding to the ideal isovolumic addition of heat, as shown by the line *ab*, Fig. 272.

The piston will then make a *third stroke*, being driven out by the high-pressure gas expanding according to the curve *bc*, which may be considered an adiabatic.

In the ideal case heat would be given to the cold body according to process *cd*, while the volume remained constant, but in the actual case the exhaust valve *E*, in Fig. 271, is opened, allowing the high-pressure gas to expand into the atmosphere until the pressure in the cylinder falls to *d*.

During the *fourth stroke* the returning piston expells the remaining gas according to the line *de*, and at *e* the starting conditions are restored, with the clearance space filled with burnt gases at atmospheric pressure.

(c) Although four strokes are required to complete the practical cycle, the work area under the line *ed* cancels that under *de;* thus the ultimate result is the development of a cycle inclosing the work area *abcda*, exactly as in the ideal Otto engine discussed in Section 56, page 94.

The two strokes corresponding to *ed* and *de* are really *pumping strokes*, used to draw in the new charge of combustible and to expel the burnt gases. They are, therefore, necessitated by practical considerations, though not essential to the ideal cycle.

(d) A real engine of this type is shown semi-diagrammatically in Fig. 273. The cylinder head has been broken away to show the valves, which correspond exactly to valves *I* and *E* of Fig. 271. Instead of using a mixture reservoir, assumed in the ideal case, the real engine forms its own mixture during the suction stroke, drawing the constituents through the pipes in the figure.

The cylinder and cylinder head of the real engine are water-jacketed to prevent overheating of the metal.

The valves in this case are positively operated by linkage

(not shown) moved by cams on the " half-time shaft," or " cam shaft," shown along the side of the engine. This shaft is driven by gears from the crank shaft, the gears being so proportioned as to give the cam shaft one revolution for every two revolutions of the crank shaft.

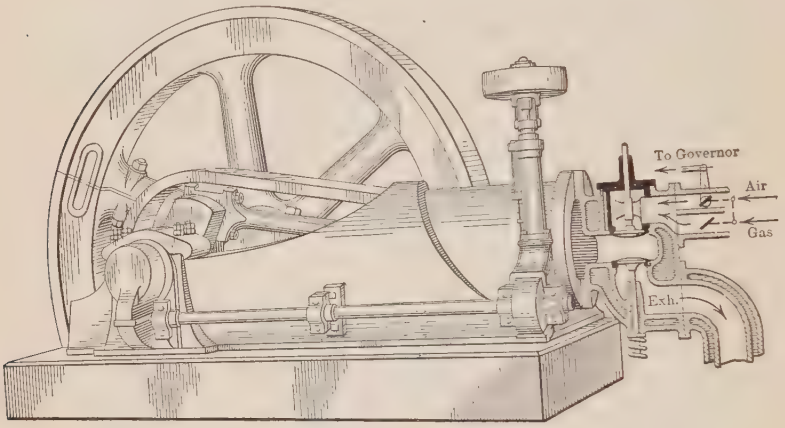


Fig. 273.

198. The Air Card. (a) The series of operations just described cannot be carried out perfectly in any real engine; thus the picture of what actually occurs in the working cylinder is quite different from Fig. 272.

(b) The losses in the cylinder are commonly determined by comparing the actual diagram with the diagram of an ideal Otto cycle with air as the working substance. This ideal diagram is also called the " *air card*," or " *air diagram*," and it is constructed for an engine like that shown in Fig. 271, operating as described in the last section, but with air only in the cylinder.

Referring to Fig. 272, it is assumed that at the point d the clearance and displacement volumes are filled with air at atmospheric pressure and temperature; that the compression da is adiabatic; that at a heat is added equal to that which would be liberated by complete burning of the combustible mixture used per cycle in the real engine; that from b the expansion is adiabatic to c; and that the heat is then removed, as in the ideal case, until the air returns to starting conditions at d.

(c) The pressure at a can be found from Eq. (45b) and the temperature can then be computed from Eqs. (51) and (52).

The height of the point b is obtained thus: First find the theoretical temperature to which this quantity of heat would raise the charge of air, with heating taking place at constant volume, and with specific heat of air constant; then determine the corresponding pressure P_b from the relation $\frac{P_a V_a}{T_a} = \frac{P_b V_b}{T_b}$.

199. Real Indicator Card for Four-Stroke Cycle. (a) In Fig. 273 is shown a real engine with the cylinder surrounded by a *water jacket* to prevent overheating of the metallic walls. Fig. 274 shows another engine in which the cylinder is covered by ribs presenting large radiating surface so that *air* may be the *cooling* medium instead of water. The actual cards obtained from such engines differ in many respects from the ideal air card just discussed, because of (1) chemical and physical properties of the real working substances; (2) thermal properties of the metallic parts of the engine; and (3) mechanical faults, such as leaking piston and valves. The variations are shown in Fig. 275, in which the real card (full lines) has been superimposed on the ideal diagram (dotted). Parts of the real card have here been overdrawn to accentuate the variations.

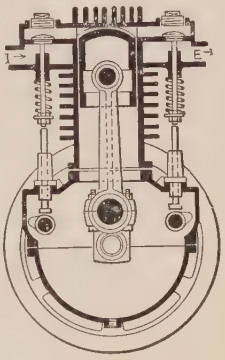


Fig. 274.

(b) Starting at the end c of the expansion line, in the ideal case with the mechanism of Fig. 271, the exhaust valve would be opened to allow the charge of the preceding cycle to escape into the exhaust pipe. In the real case, however, this valve must start to open before the end of the stroke, say at c', which is usually at from 85 to 90 per cent of the out-stroke. This is necessary so that the valve, which cannot be opened instantly to its full extent, may have time to open fully before the end of the stroke is reached; and because the gas in the cylinder, due to its inertia, takes an appreciable time to pass through the exhaust valve despite the fact that the gas pressure of from 15 to 35 pounds or more above the atmosphere is available to accelerate it.

From c' the expansion line drops rapidly to the end of the stroke, both because additional space is vacated by the piston

as it continues outward, and because of the exit of gas from the cylinder.

(c) The line $d'e'$ is higher than the ideal exhaust line de. This is due to the pressure difference necessary to cause the flow of gas through the exhaust valve and pipe to the atmosphere. As the area opened by the valve is limited by practical considerations, a high average velocity of gas flow through this valve is

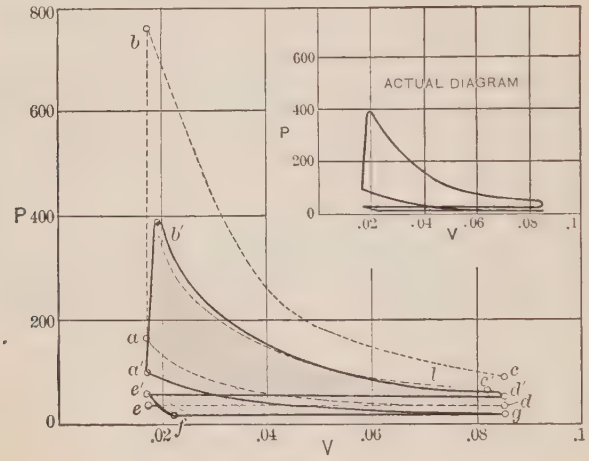


Fig. 275.

necessary in order to empty the cylinder in the available time. This velocity varies from 80 to 125 feet or more per second, and to produce it the exhaust pressure line $d'e'$ must be from one to three pounds above atmospheric. Instead of being straight, this line is generally more or less wavy because of the inertia of the gases.

(d) At e', with the piston at the end of the stroke, the clearance is filled with products of combustion at a pressure slightly above atmospheric and at a temperature probably 700° to 900° Fahr. As the piston starts on the "suction stroke," these gases expand to some pressure f, from one to six pounds below that of the mixture supply (which is usually at atmospheric pressure), before the new charge begins to flow through the open inlet valve into the cylinder. This flow continues as the piston moves out until the end of the stroke is reached at g, when the cylinder is filled with a mixture of the new charge and the

burnt gas previously left in the clearance. This "*suction line*," *fg*, is only approximately straight and horizontal.

Evidently during both the exhaust and the suction strokes the piston must *do work on the gas*, and this decreases the power that the engine can deliver.

(e) The compression line ga' is generally below da (1) because compression begins at g with pressure below atmospheric; (2) because the physical properties (γ, etc.) of the real mixture are different from those assumed for air (3) because the process is not adiabatic, for there is heat interchange between the gas and the walls of the piston, cylinder, and head; and (4) because of leakage past piston and valves. This process is generally intermediate between an adiabatic and an isothermal.

(f) At or near a' ignition occurs, and as it actually takes an appreciable time for the flame to spread throughout the mixture, and as the piston does not remain stationary at the end of the stroke during the complete process of combustion, the sloping ignition line $a'b'$ results, instead of the vertical line ab of the ideal process. Combustion is seldom complete, even when the highest pressure is reached, hence heat is still being added when expansion starts.

(g) The pressure does not rise as high as the ideal value b, presumably because (1) the initial pressure a' is less than the ideal at a; (2) the movement of the piston increases the volume during combustion; (3) the average specific heat of the mixture is different from that assumed for air and increases as the temperature rises; (4) the surrounding metallic walls absorb some of the heat generated; (5) the chemical reactions accompanying combustion may result in products occupying less volume than the original mixture; (6) there may be a certain amount of dissociation at the higher temperatures; and (7) there may be leakage past the piston and through the valves.

(h) The expansion line is at first generally above an ideal adiabatic curve $b'l$ because of "*after burning*," or the continuation of combustion, which usually adds heat in excess of that absorbed by the metal walls and that converted into external work. Later, as the motion of the piston continues, the relatively cooler cylinder walls are uncovered and they rapidly absorb heat from the gas, causing the expansion line to drop below the adiabatic.

(i) During part of the compression, and all of the combustion and expansion, *heat is absorbed by the inclosing metallic walls*, from which part of it is carried away by the water or air jacket. This is a direct loss, but it is necessary in order to prevent overheating the metal.

(j) During the suction stroke the incoming gas receives heat from the confining walls and from the exhaust gas still remaining in the clearance space, until at the end of the stroke the gas and inner surface of the walls are probably at nearly the same temperature. Because of the expansion of the gas due to this temperature (which is often from 700° to 900° Fahr.), and because of the reduction of pressure below atmospheric during the suction stroke, the *weight of fresh mixture* drawn in *is reduced*, and hence less than the theoretical work per cycle is done in a given cylinder.

200. Losses in the Four-Stroke-Cycle Engine. (a) A complete analysis of all the losses in the cylinder of an internal-combustion engine would be very complicated, and is as yet unsatisfactory because of the lack of experimental data. For the purposes of this book, it will serve to indicate the principal sources of loss and to treat them qualitatively rather than quantitatively.

(b) The **Otto cycle efficiency** is from Eq. (81)

$$Ef = 1 - \left(\frac{V_{a'}}{V_g}\right)^{\gamma-1}.$$

It is seen to be dependent only on the compression ratio ($V_{a'}/V_g$), which the designer can control, subject to practical considerations, by the selection of proper clearance volume, $V_{a'}$. Thus theoretically, nothing is lost by the low pressure or high temperature at g.

(c) Section 199 (j), however, showed that the actual weight of fresh charge drawn into the cylinder during the suction stroke is always less than the theoretical, and this of course reduces the power developed.

Let $V_{s'}$ be the volume corresponding to the actual weight of gas drawn in, and V_s be that equivalent to the piston displacement per stroke, both volumes being measured at atmospheric pressure and temperature. Then the ratio $\left(\frac{V_{s'}}{V_s}\right)$ is called the

volumetric efficiency. In practice its value may reach 90 per cent in well-designed slow-speed engines, or it may be reduced by high speed or incorrect design to 50 per cent or less. Evidently, in a given engine the amount of heat liberated per cycle depends on the volumetric efficiency, and hence for definite power output a *lower volumetric efficiency makes necessary larger cylinder* and greater cost of engine unless operated so as to give more cycles per minute.

(**d**) The effect of the falling of the **real compression line** below the adiabatic, upon the performance and efficiency of the engine, is difficult to state in any general way. Since the line lies between the isothermal and adiabatic, the work done is slightly less than that which corresponds to an adiabatic process, and this compensates more or less fully for the loss of heat which makes this line fall below the adiabatic, and for the corresponding lowering of the efficiency.

(**e**) The **combustion line** $a'b'$ represents the most complicated process in the cycle and is the most difficult to investigate, as the phenomena take place with comparative rapidity and vary with the character of the mixture, the method of ignition, the surface form of the combustion space, etc. The real loss during this process cannot be accurately measured by comparison with the ideal air diagram, but could be determined by comparison with a card drawn for the working substance actually used in the real engine, considering specific heats variable and accounting for any other theoretical modifying conditions. As this would mean a different standard for every fuel, and for every different mixture of fuel and air, the "air standard" is retained for simplicity.

In considering the sloping combustion line $a'b'$, it is again a case of balancing gains and losses. The piston movement reduces the maximum pressure and temperature, thus decreasing the heat lost to the cylinder walls, but this is offset more or less completely by the larger surfaces exposed while the temperature is high. The slope which will give the highest efficiency cannot be predicted, but usually an inclination which will bring the top of the combustion line at about 2 per cent of the stroke seems to give the best results. The loss of area between the real and theoretical combustion lines is partly compensated by the broadening of the top of the diagram. This change in form of the

diagram improves the mechanical operation of the engine because the pressure changes are less sudden and less intense.

(f) The **expansion line** $b'c'$ generally incloses slightly more area than the adiabatic $b'l$ (Fig. 275), unless the engine is of such proportions as to expose excessive wall area.

By opening the exhaust valve at c', as in Fig. 276 (a), less area of diagram is usually lost than if the opening is at the end of

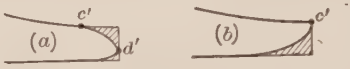


Fig. 276.

the stroke, as in Fig. 276 (b); and as less hot gas remains in the cylinder the tendency to overheat the metal walls is reduced.

The actual heat interchanges during exhaust are problematical. The enormous rush of gas through the exhaust valve consumes heat which is lost to the atmosphere, but there is a corresponding gain due to increased volumetric efficiency resulting from contact of the new charge with cooler walls.

Quite remarkable success has been achieved by engines in which cold air is blown through the cylinder during part of the exhaust period. This operation cools the walls and tends to remove the burnt gases from the clearance space, and hence the charge drawn in is cooler and purer than in the ordinary type of engine. Such engines are known as "*scavenging*" or "*positive scavenging*" engines.

201. Requirements for High Efficiency of Combustion. (a) There are two antagonistic requirements for high efficiency of combustion: (1) *The final compression pressure and temperature* (at a', Fig. 275) *must be high*, since this not only gives high efficiency theoretically (see Eq. (80)), but also because experience shows that the charge is often more readily ignited and burned from high pressure. The limit is reached when the pressure is so high as to cause "*preignition*," that is, spontaneous ignition of the mixture during compression. With other things equal, the greater the pressure and temperature at the end of compression, the higher will be the final temperature at b'. (2) *The maximum temperature attained* (at the point b') *should be as low as possible*, because the specific heats and loss of heat to metallic walls increase rapidly at high temperatures.

(b) These two requirements for high actual efficiency can be harmonized in practice by using a mixture with large excess of air. This may be highly compressed without danger of preignition; it burns rapidly enough at high pressures for satisfactory combustion; and, because of the excess of air present, the final temperature attained is comparatively low.

Unfortunately, however, the mixture which gives highest actual efficiency "on the brake" does not give maximum possible power from a cylinder of given size operating at given speed; thus there is a tendency to operate engines with mixtures "richer" in combustible than those giving the highest efficiency.

202. Indicated Work and Power of the Four-Stroke-Cycle Engine. (a) In the diagram shown in Fig. 277, with the "lower loop" *fghef* considerably exaggerated, the arrows indicate the directions in which the various lines are traced.

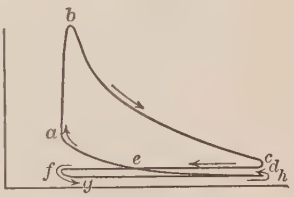


Fig. 277.

(b) If areas on a PV-diagram surrounded by lines generated in one direction (here clockwise) represent work done *upon* the piston, or positive work, then areas inclosed by lines of reverse direction (here counter-clockwise) indicate work done *by* the piston upon the working substance, or negative work.

Thus the work represented by the upper area or "loop," *abcdea*, is positive; the work corresponding to the "lower loop," *fghef*, is negative; and the *net useful work on the piston* would be represented by the *difference* between these areas.

(c) The exact interpretation of **"indicated power"** in the case of a four-stroke-cycle gas engine is still unsettled. All things considered, it seems best to calculate i.h.p. from the upper loop alone. Then the difference between the i.h.p. and the d.h.p. is the work lost in overcoming both the *fluid friction* and the *friction of the mechanism.*

The **fluid-friction loss** is measured by the area of the lower loop; it would equal zero with frictionless flow. The engine-friction loss is the difference between the total friction loss and that due to fluid friction; it would equal zero with a frictionless mechanism.

(d) The **mechanical efficiency** is the ratio $\frac{\text{d.h.p.}}{\text{i.h.p.}}$. As applied to gas engines, it includes both kinds of friction loss when the

i.h.p. is computed according to the method just given. It is advisable to adhere to this method because of the difficulty of obtaining the correct area of the lower loop.

203. The Two-Stroke-Cycle Otto Engine. (a) Comparison of single-cylinder single-acting Otto engines of the four-stroke-cycle and the two-stroke-cycle types shows that in the former there is one power stroke out of four, while in the latter there is one power stroke out of two. Hence with the same rotative speed and cylinder dimensions the two-stroke-cycle engine theoretically should give twice the power of the four-stroke-cycle engine, and should require much less flywheel weight to maintain the same degree of uniformity in rotative speed.

Moreover, in the four-stroke-cycle engine the mechanism, which is designed for very high pressures, is used half the time for pumping gas at low pressure (while forming the lower loop of the diagram). And to make matters still worse, the density of the mixture, and therefore the weight of gas drawn in per cycle, is reduced by heat received from the hot cylinder walls, and this increases the cylinder size for a given power output. In the two-stroke-cycle engine, on the other hand, a separate, specially designed pump, with cool walls, may be used more effectively for this service.

(b) The two-stroke-cycle engine is represented diagrammatically in Fig. 278. The pump cylinder has an inlet valve *A*, and

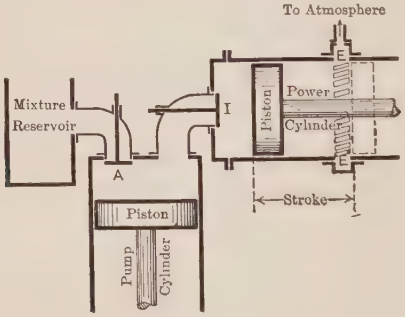


Fig. 278.

a discharge valve *I*, which latter also serves as an inlet valve to the power cylinder. This cylinder has a ring of ports *E*, cut through the walls at such a point that the piston, by uncovering them near the end of its stroke, acts as an exhaust valve.

(c) Now imagine the **ideal cycle** performed without mechanical or thermal loss as follows: Consider the power cylinder filled with mixture at atmospheric pressure, the power piston just covering the exhaust ports *E*. The *first stroke* is to the left, causing compression of the charge according to the line *d'a* in Fig. 279 (*a*). Combustion produces the line *ab*. Expansion during the *second stroke* occurs according to the line *bc*; and when the piston passes the ports, *E*, exhaust occurs according to the line *cd*, thus nearly completing the cycle in the power cylinder. Meanwhile the pump piston has moved down and drawn in from the reservoir a charge of mixture sufficient to fill the power cylinder, the theoretical process being represented by *ef* in Fig. 279 (*b*). The valve *A* is then closed, and after the pressure in the power cylinder has dropped to atmospheric at *d* in Fig. 279 (*a*), the valve *I* is opened and the pump piston is quickly raised, driving the mixture into the power cylinder, according to the theoretical line *fe*. While this is occurring the power piston moves from *d* to *d'*.

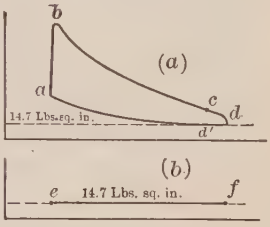


Fig. 279.

In the ideal case the charge entering the power cylinder will drive the remaining exhaust gases out through the ports as it moves down the length of the cylinder in a solid column, and arrive at the exhaust ports just as the returning power piston covers them. The power cylinder is thus charged with a combustible mixture, with volume as shown at *d'*, at atmospheric pressure, and with the conditions assumed at starting.

The theoretical pumping work, represented by the area under *ef* minus the area under *fe*, is zero, as in the case of the ideal four-stroke cycle. In the power cylinder the Otto cycle is followed except at the end *cdd'*, which is modified for practical reasons.

(d) The differences between **the actual-work diagram** and the ideal Otto cycle are quite similar to those occurring in the four-stroke-cycle engine, and arise largely from the same causes.

The pump does not actually operate in the ideal manner. It is usually driven from a crank on the engine shaft, and in consequence the gas must be pumped to some intermediate reservoir, where it must be maintained at a pressure of from 0.5 to 7 pounds

above atmospheric in order to fill the power cylinder in the short time available after the inlet valve opens. Energy is not only lost in overcoming the pump friction and resistance to flow, but is also expended in compressing the mixture in the pump cylinder. The work done on the gas is shown by the area of the actual pump card.

Because of its great velocity, the entering charge generally mixes more or less with the burnt gases, and some portion usually escapes through the exhaust ports before they are covered.

Although theoretically the two-stroke-cycle engine would develop twice the power given by a four-stroke-cycle engine of the same size and r.p.m., the actual ratio is usually from 1.4 to 1.6, owing to the losses due to the method of operation.

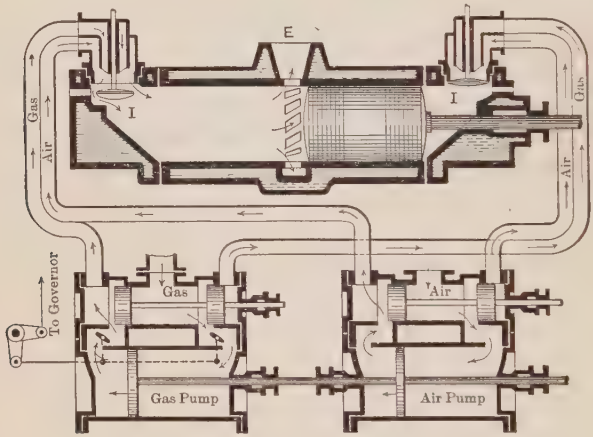


Fig. 280.

(e) In some two-stroke-cycle engines the power cylinder is first scavenged by admitting air under pressure ahead of the mixture so that none of the fresh charge escapes with the exhaust. The saving thus effected is, however, offset more or less completely by the necessity of using two pumps instead of one, with increased complexity and greater expenditure of energy in pumping. One engine of this type is shown semi-diagrammatically in Fig. 280 and is known as the *Koerting* design.

In some single-acting engines operating on the two-stroke cycle, the mixture is first admitted to the crank case, as in

Fig. 281 (*a*), where it is compressed by the under side of the piston acting as a pump during the down stroke. The opening of separate inlet and exhaust valves is replaced by the uncovering of the inlet and exhaust ports by the piston when near the end of its stroke, as shown in Fig. 281 (*b*). The fresh charge entering through the inlet port is so baffled as to assist in driving the burnt gases toward the exhaust port.

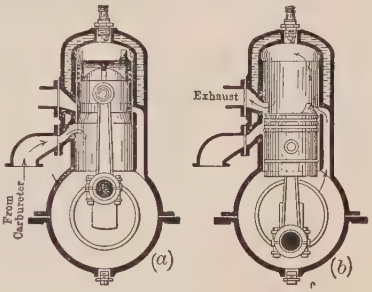


Fig. 281.

204. The Diesel Engine. (**a**) Engines commercially known by this name operate approximately on the cycle discussed in Section 58, and shown in Fig. 29. The real cycle may be completed in either two or in four strokes.

(**b**) The *mechanical operations* within the power cylinder of the real engines are very similar to those of the Otto engine. With four-stroke operation the *suction stroke* charges the cylinder with air, which on the *return stroke* is compressed into a clearance volume so small that the terminal pressure is very high, equal to 400 to 500, or more, pounds per square inch, with correspondingly high temperature. Just before, or when, the piston reaches the end of the compression stroke, a small quantity of finely atomized liquid fuel is blown into the clearance space by means of air at very high pressure. The fuel immediately ignites, due to the high temperature of the air that was compressed in the clearance space by the engine piston. The combustion which ensues continues a little longer than the period of injection. As the moving piston increases the volume a little faster than the gas tends to expand under the action of the heat developed, and as heat is lost in the cylinder walls, the upper line of the card slopes slightly, as in Fig. 282. In this figure the ideal and real diagrams are shown superimposed, with the lower loop exaggerated.

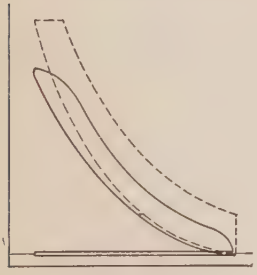


Fig. 282.

(c) Within the past few years several designs of two-stroke-cycle engines operating on this cycle have been made, and some of these give considerable promise of success.

205. Modifications to Suit Different Fuels. Theoretically, the internal combustion engines just discussed can use any fuel that can be introduced as gas or vapor (or even as finely divided solid) in a combustible mixture. In practice the fuels used are the combustible commercial gases, petroleum products, the by-product tars from gas works and such, and alcohol. It is generally necessary to make the design of some parts of the engines and auxiliaries special for each different fuel, and as a result commercial engines are often named after the fuel used; thus there are "producer-gas engines," "gasoline engines," "kerosene engines," etc. The chief differences between types are given in the following sections.

206. Compression and Maximum Pressures. (a) In practice one of the most important considerations is the *final compression pressure*. In theory the thermal efficiency will increase with the final compression pressure, and within limits this is true in the real engines (see Section 219).

It is found, however, that compression above certain limiting pressures causes *spontaneous ignition*, or *preignition*, which tends to stop the engine.

With some fuels the spontaneous-ignition temperature and pressure are so high that the compression limit is not set by preignition, but by commercial considerations. Thus, with very high compression the "up-keep" may exceed the gain due to increased thermal efficiency. For example, engines using blast-furnace gas usually compress only to about 175 pounds gauge, or even less, although the preignition pressure is much higher, and the thermal efficiency of engines compressing to 200 pounds has been shown to be better.

(b) The **usual compression pressures** (terminal) in the different types of engines as now designed are given in Table XI. This shows that the lowest compression pressures are used with the fuels high in hydrocarbons, while high pressures are used with fuels low in these constituents.

High compression increases the thermal efficiency, not only

because it improves the theoretical cycle, but also because it aids ignition and makes combustion more rapid. This is particularly true with the weaker fuels, like blast-furnace gas.

TABLE XI. — COMMON COMPRESSION PRESSURES.

Fuel.	Comp. Press. Lbs. above Atmos.	Fuel.	Comp. Press. Lbs. above Atmos.
Kerosene	50 to 75	Producer gas	120 to 150
Gasoline	60 to 75	Blast-furnace gas	140 to 175
Illuminating gas	70 to 90	Alcohol	140 to 180
Natural gas	100 to 125		

(c) In theory, with other things equal, the greater the calorific value of a charge and the higher the temperature before ignition, the higher will be the maximum temperature and pressure attained by the combustion. In practice this is modified by the considerations brought out in preceding sections of this chapter.

In general, engines in which the maximum pressure is high because of rich mixtures and high compression must be stronger and heavier than those using "weak" mixtures and low compressions.

CHAPTER XXV.

INTERNAL-COMBUSTION ENGINES (*cont.*).

Mechanical Features.

207. Cylinder Arrangement. (a) In the theoretical discussion of preceding chapters, only single-cylinder, single-acting engines were considered. In practice there are three principal reasons for making "*multi-cylinder units*" and for making "*double-acting engines.*" These are:

(1) The *turning effort* at the shaft of an internal-combustion engine with one single-acting cylinder is very uneven. This can be partly counteracted by the use of a very heavy flywheel, but this is objectionable for several reasons. As a result, both multi-cylinder and double-acting constructions are used to give overlapping cycles and therefore more even turning efforts.

(2) The *power* which can be obtained from a given cylinder depends upon the quantity of heat which can be liberated in that cylinder by combustion. This, in turn, depends upon the volume of mixture which can be contained in the cylinder, and hence upon the cylinder dimensions. Experience has shown that a cylinder diameter of from 42 to 45 inches is about as large as is commercially advisable with present methods of construction and with the materials now in use. As the length of the cylinder cannot be increased without limit, it results that the *maximum power* that can be obtained *per cylinder is limited.*

A single-acting cylinder can be constructed to develop from 500 to 700 horse power, and when larger powers are required per unit, double-acting or multi-cylinder constructions must be used.

(3) Because of the high pressure generated in gas-engine cylinders, the forces transmitted by the moving parts of the engine are very large, and these parts must be made correspondingly large. With the single-cylinder construction, the *unbalanced forces* are of *great magnitude.* These can be decreased by a proper arrangement of several cylinders.

(b) The attainment of a more even turning effort than is possible with a single cylinder is of such great importance that very few single-cylinder engines are now built in sizes above 50 horse power, and they are seldom used in sizes above about 25 to 30 horse power excepting for work where close regulation is not very important.

An idea of the handicap under which internal combustion labors in this respect can best be obtained by a comparison with a single-cylinder double-acting steam engine. To produce as many impulses in a given time as an engine of this type, a single-acting four-stroke-cycle engine, running at the same speed, would require four cylinders; if double-acting, two cylinders would be needed. The two-stroke cycle construction gives the same number of impulses as does a steam engine of the same type, i.e., single- or double-acting.

The double-acting, internal-combustion engine, however, offers more difficulties in construction and operation than does a similar steam engine. The piston and piston rod must be water-cooled in order to prevent overheating, and the maintenance of a tight piston-rod packing is more difficult with hot gases than with steam.

208. Classification. (a) Like steam engines, the internal-combustion engines are classified in a number of ways. The principal designations, and a brief discussion of each, are given in the following paragraphs. Such things as center-crank and side-crank construction, and right- and left-hand arrangement are common to all kinds of engines and need not be further considered.

(b) Internal-combustion engines are made both *vertical* and *horizontal*. For sizes up to about 500 to 700 horse power either construction is used, each having certain advantages and certain disadvantages. Above 700 horse-power commercial economy generally dictates double-acting cylinders. Very few vertical engines have been built double-acting, as there is considerable difficulty in accommodating the valves for the lower cylinder end in this construction, hence the larger powers are nearly always supplied by horizontal engines. The vertical engine has the advantage of occupying much less floor space than the horizontal and can be mounted on a less massive foundation. It is generally operated at a higher speed, particularly in the larger

sizes, and is usually built with an inclosed crank case so that lubrication can be somewhat simplified.

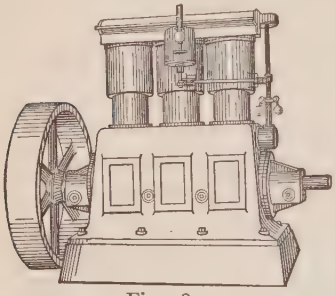


Fig. 283.

(c) The cylinders of multi-cylinder vertical engines are practically always arranged side by side and as close to one another as possible. These engines are designated as *two-cylinder vertical*, *three-cylinder vertical*, etc. A three-cylinder vertical engine is shown in Figs. 283 and 284. With horizontal engines, however, the cylinders are often widely separated, giving what is called a *twin engine.*

It is also very common practice to place two horizontal cylinders with their axes coinciding. When so arranged, the engine is called a *tandem.* A combination known as a *twin-tandem double-acting* is shown in Fig. 285.

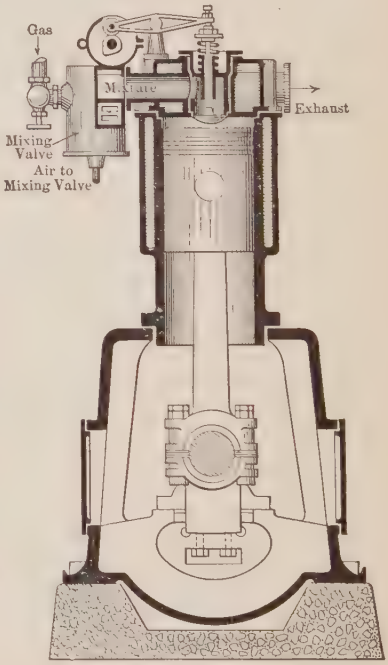


Fig. 284.

(d) Internal-combustion engines are sometimes classified according to the *use* to which they are put. Thus there are stationary engines, stationary electric lighting engines, marine engines, automobile engines, etc. From this classification has sprung another, an engine of one type being designated by its type name, even when used for a different purpose. There are thus "auto-type marine engines" and "marine-type stationary engines."

(e) Since certain modifications, or different fittings, are necessary with different fuels, internal-combustion engines are sometimes classified according to the *fuel* which they are intended

to use. Thus there are kerosene engines, gasoline engines, illuminating-gas engines, producer-gas engines, etc.

(f) These engines are also occasionally classified on the basis of the type of *governing* used (see sec. 212). Thus there are hit-and-miss engines, throttling engines, etc.

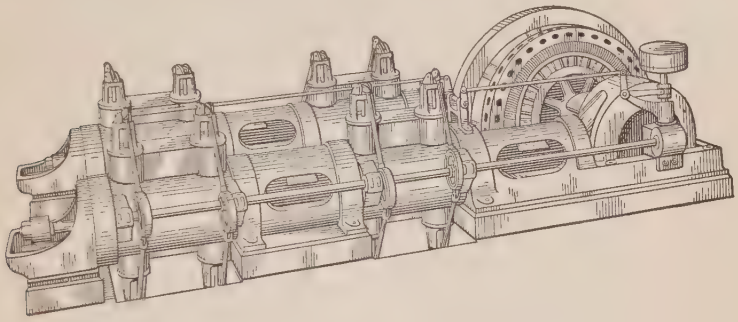


Fig. 285.

209. Methods of Producing Combustible Mixtures. (a) With fuels initially *gaseous*, a "**mixing valve**" is generally used to control the proportions of fuel and air, the two gases being made to mix intimately either before or during entrance to the cylinder. This mixing valve may be incorporated with the inlet valve or it may be separate and at some distance from it. Examples of both types are given later.

(b) Fuels initially *liquid* must either be atomized or vaporized and mixed with air to support combustion. With the more volatile liquid fuels, such as gasoline and alcohol, the process generally takes place outside of the engine cylinder in a "carburetor"; the mixture then passes to the cylinder as in the case of fuel initially gaseous. With the less volatile liquid fuels, like kerosene or crude oil, vaporization and mixing are more difficult, and generally take place within the engine cylinder, the fuel being sprayed in either by pump or air pressure and being **vaporized by heat** from hot walls or gas. Examples of such devices are given later.

210. Carburetors. (a) When an engine uses a *volatile liquid fuel*, like gasoline or alcohol, it is customary to mix the fuel vapor and air outside of the cylinder in a carburetor, in which air, which may or may not be previously heated, is brought into

intimate contact with the liquid and becomes charged with the vapor.

(b) A great variety of types of carburetors has been perfected and used. Thus there are *bubbling carburetors*, in which some or all of the air is made to pass or bubble through the volatile liquid, on its way to the engine. There are *surface carburetors*, in which the volatile liquid is spread over screens, marbles, or anything else which will give a large wetted surface over which the air may be drawn. *Wick carburetors* have also been used. In these the liquid is drawn up into wicks by capillary action, and the air passing over the surface of the wicks vaporizes part of the exposed liquid.

(c) Practically the only type now used in this country is the *jet carburetor*. This apparatus is made in many forms, but the fundamental principle of all is the same. A fine jet of gasoline is injected into the air pipe and generally only during the suction stroke. The impelling force is usually either the pressure due to a slight head of gasoline or the difference between suction pressure and atmospheric pressure, or this difference augmented by the suction effect of rapidly moving air upon a nozzle immersed in it.

(d) One of the most common types of jet carburetor, known as a *carbureting valve*, is shown in Fig. 286. The valve is sometimes the inlet valve of the cylinder, but more often it is a separate valve through which air is admitted to the mixture pipe leading to the main inlet valve. A small hole is drilled in the seat of the carbureting valve in such a position that it is closed when the latter is seated. When the valve rises (automatically) to admit air to the engine, the liquid under slight pressure issues from the hole in a very small stream, which mixes with the air and is partly or wholly vaporized before the mixture enters the cylinder.

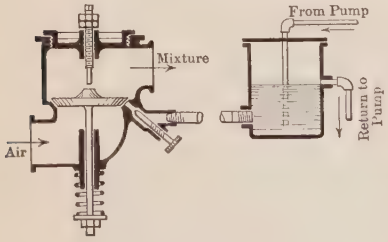


Fig. 286.

(e) Another form of jet carburetor is shown in Fig. 287. The liquid is maintained at such a height that its surface almost reaches the tip of the spray or injection nozzle when quiescent.

The air passing around this nozzle on its way to the engine creates a partial vacuum at the nozzle, which vacuum augments the lowering of the pressure caused by suction in the engine. The air pressure on the surface of the liquid in the small tank then forces a fine jet out of the nozzle, and this is picked up by the surrounding air. The throat, or *Venturi tube*, increases the velocity of the air flowing through it, which materially assists in picking up and carrying the liquid during vaporization. This type is commonly used on stationary engines, the liquid level being maintained by a direct-connected pump and overflow as shown.

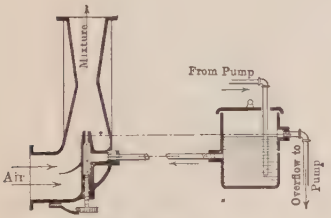


Fig. 287.

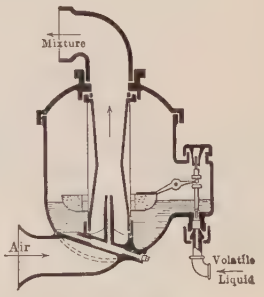


Fig. 288.

(f) In Fig. 288 is shown a type of *float-feed carburetor*. This is similar in action to that last described, the principal difference being the float for maintaining the proper liquid level. This operates by opening and closing the small valve shown as the liquid level sinks and rises, the liquid being supplied to this valve under pressure.

This type of carburetor is most common on automobile and marine engines, the central float, which maintains approximately the correct level despite tipping of the carburetor, and the compact structure both recommending it for such purposes.

(g) When an engine is run at widely *varying speeds*, it is a very difficult matter to adjust a carburetor of the type last shown to give a suitable mixture under all conditions. If the mixture is correct at low speeds, it is apt to be too rich at high speeds. This is overcome by introducing an *auxiliary air valve* between the spray nozzle and the engine This valve, operating automatically or under hand control, admits air, which, combining with the over-rich mixture, forms one of correct proportions.

Similar devices are sometimes necessary with the carburetor used on engines which run at a constant speed, being used to make the adjustment of the carburetor easier or more certain.

211. Treatment of Heavy Oils. (a) The use of fuels like kerosene, distillate, crude oil, and such, presents greater difficulty than the utilization of gasoline or alcohol. Kerosene can be handled more or less satisfactorily with carburetors similar to those described, but, being less volatile than gasoline, the action

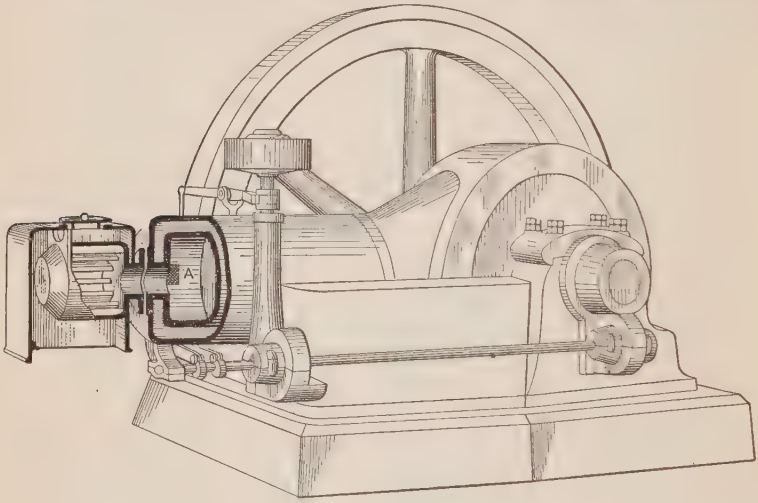


Fig. 289.

is not so perfect. It is generally necessary to preheat the air and to jacket the mixture pipe with hot jacket water, or with exhaust gases. Even with these additions, it is often found difficult to operate satisfactorily, and most carbureting kerosene engines are arranged to spray water into the cylinder or to saturate the mixture with water vapor on its way to the cylinder, particularly when running under heavy loads. Just what the action of the water vapor may be is still undetermined, but it seems to give more certain, quieter, and cleaner combustion.

(b) Many kerosene and other oil engines operate on what is known as the *hot-bulb* or *hot-head* principle. An engine of this type is shown in Fig. 289.

The oil is injected into the hot bulb during the suction or compression strokes and is there vaporized by the hot walls. Air

is compressed into the bulb during the compression stroke of the engine, and, when the mixture acquires the proper proportions, spontaneous ignition takes place. The bulb is heated to redness by a blow-torch before starting the engine, and thereafter is maintained at the proper temperature by the heat generated during combustion.

There is always a certain amount of carbon or lampblack deposited within the hot bulb by the "*cracking*" of the oil molecules during vaporization, and it is therefore necessary to clean the bulb periodically.

(c) Practically the only other distinct method of using the heavier oil fuels in internal-combustion engines is that exemplified in the Diesel oil engine described in Section 204. This gives by far the most perfect combustion with the heavier fuels, but is open to criticism because of the high pressures involved.

(d) To overcome this difficulty, engines are now being built which may be considered a compromise between the hot-bulb and the Diesel types. The pressures are lower, but the hot bulb insures successful ignition and combustion. These engines are proving highly economical in the use of fuel, and can be kept in good mechanical condition with greater ease than can the high-pressure Diesel engine.

212. Methods of Governing Internal-Combustion Engines. (a) Stationary engines are generally mechanically regulated to maintain *approximately constant speed* of rotation. Automobile and marine engines are commonly hand-governed, although they are sometimes fitted with a limit governor to prevent overspeeding, or "*racing.*"

(b) In order to govern or regulate an engine, the i.h.p. must be varied to suit the demand, as shown in Section 134. There are *three* available methods of doing this: (1) The amount of energy made available per cycle may remain constant, but the number of cycles per unit of time may be changed; (2) the number of cycles may remain constant and the amount of energy made available per cycle may be varied; and (3) a combination of the two preceding may be used.

(c) In general, there are four different ways of applying these methods. They are called: (1) hit-and-miss governing, (2) quantity governing, (3) quality governing, and (4) combination sys-

tems. These are each considered in detail in the following paragraphs.

(d) In **hit-and-miss governing**, the number of working cycles per unit of time is varied so as to adjust the average i.h.p. to the demand for power. With this system, some part of the mechanism for opening the inlet valve is under the control of the governor, so that when a "*working cycle*" is to occur it *hits* another part and opens the valve, but when the cycle is to be omitted it *misses* engagement and the valve remains closed. When a miss occurs, not only does the inlet valve remain closed, but the exhaust valve is usually held open, so that, during the strokes corresponding to the ordinary cycle, the piston pumps exhaust gas into and out of the exhaust pipe without waste of energy, except for the slight friction and heat loss.

In some engines, when the working cycle is to be omitted, a fuel valve, which is separate, is held closed while the inlet and exhaust valves act as usual; thus the piston draws in a charge of pure air, which it compresses, expands, and exhausts. This method is generally considered less satisfactory than the former, because of the cooling effect on the cylinder walls.

With hit-and-miss governing all working cycles are theoretically exactly alike, and are equal to the maximum for the particular engine. As all types of internal-combustion engines show greatest thermal efficiency when developing normal cycles of about maximum power, this method of governing has the theoretical advantage of giving *high thermal efficiencies at all loads*. The cycles actually produced, however, are not all alike, because of irregular cooling and heating effects, the varying mixtures resulting from intermittent operation, etc. The variations become more marked with increase of the number of misses, and the method therefore gives lower efficiencies at light loads than would be expected. In general, however, it is the most economical method of governing yet devised. As considerable intervals of time may intervene between "working" cycles, a very heavy flywheel is needed on engines governed by this method.

Hit-and-miss governing is very satisfactory for engines where close speed regulation is not necessary, and is commonly used on the smaller sizes, say up to 25 or 50 horse power. Where close regulation is required, as for the operation of alternators in parallel, it is practically never used.

(e) In **quantity governing,** the number of cycles and the proportions of the mixture are maintained constant, but the *amount* of mixture admitted per cycle is varied to suit the power demand. This is generally done in one of two ways, — by "cut-off governing," or by "throttling governing."

In **cut-off governing**, after the amount of mixture necessary to produce the required power has been taken in, the inlet valve is closed, and the charge expanded as the out stroke, or suction stroke, continues. The cycle is then completed as usual, producing under low load a diagram like that of Fig. 290, in which the lower loop is exaggerated for clearness.

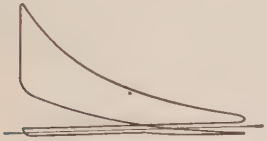


Fig. 290.

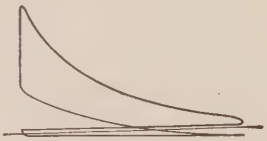


Fig. 291.

In **throttling governing,** except at the maximum load, the charge is throttled during the entire suction stroke to reduce the amount of mixture entering the cylinder. This gives a diagram like Fig. 291, in which the lower loop is again exaggerated.

In both of these methods of governing, the reduction in quantity of mixture with decrease in load is accompanied by a *lowering of the compression* curve. If not carried too far, this is desirable from a mechanical standpoint, as it tends to produce more uniform turning effort, and reduces the necessary weight of flywheel.

Of the two methods the cut-off is the better because it gives a smaller lower loop and less lost work. It also has the advantage that the governor action is delayed to the latest possible instant in the cycle, and hence each working cycle more nearly meets the power demand.

(f) In **quality governing** the number of cycles and quantity of material per cycle are maintained constant, but the proportion of gas to air, or quality of the mixture, is varied, so that the power developed in the cylinder just meets the power demand.

Since the same volume of mixture is drawn in each cycle and is compressed to the same pressure, the efficiency is theoretically

constant at all loads. In practice, however, each fuel has an air-to-gas ratio that gives best results; thus it follows that this method of regulation gives maximum efficiency only at one particular load. With some fuels it is exceedingly difficult to obtain satisfactory ignition of the very " weak " mixtures introduced at low loads, and such mixtures also burn very slowly, the combustion continuing in extreme cases throughout the entire expansion stroke.

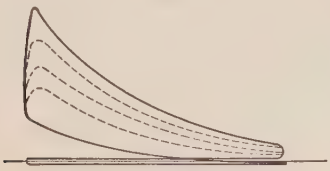


Fig. 292.

A group of indicator diagrams from a quality-governed engine is given in Fig. 292. The slow burning of the weak charges is shown by the gradual tilting of the combustion line as the load decreases.

The *constant compression pressure* has an undesirable effect on the crank effort (see (e) of this section), as the m.e.p. of the compression line does not change with the m.e.p. of the expansion line.

(g) **Combined systems** are sometimes used in an effort to obtain the advantages of the different methods previously described with as few as possible of their disadvantages. Thus hit-and-miss governing may be used at low loads and quality governing at the higher loads which call for sufficient gas to make a readily ignitable mixture. Or quality governing may be used at the higher loads, gradually merging into quantity governing as the load decreases.

All these combinations tend to complicate the valve gear and call for more or less sensitive and intricate adjustments. They are, therefore, commercially handicapped, though theoretically desirable.

(h) As the form and area of the card may be changed by *altering the time of ignition*, this might be used for governing. It is actually used for that purpose to a certain extent in marine and auto engines. Since there is some best time of ignition for each mixture in each engine running at each speed, it is generally better to change the time of ignition to suit the conditions brought about by governing rather than govern by changing the time of ignition.

In some combination systems an ignition timing device under

control of the governor has been incorporated, but it has generally been found more satisfactory to trust to hand timing.

213. Gas Valves, Mixing Valves, etc. (a) When gas is supplied an engine under pressure, as is generally the case in all except "suction gas-producer" plants (see Fig. 5), a *gas valve* of some sort is necessary to shut off the gas supply during all but the suction stroke of the engine.

(b) This valve may be combined with the inlet valve of the engine, giving the arrangement shown diagrammatically in Fig. 293. The *air* and *gas cocks* shown are used for proportioning the mixture by hand, and the gas cock is also used as a permanent shut-off valve. Such an arrangement can be used with hit-and-miss or with quantity governing, but is obviously unsuited for quality governing because of the hand regulation of the proportions.

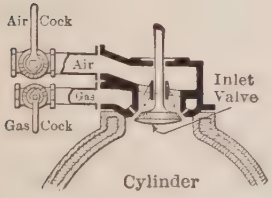


Fig. 293.

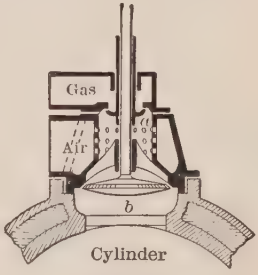


Fig. 294.

(c) The gas valve is more commonly a separate valve, although it may be carried loosely on the same stem as the inlet valve, as *a* in Fig. 294. When thus made separate from the inlet valve, it can be put under governor control, so that any kind of governing can be adopted, at the option of the designer. In all cases it is common practice to supply gas and air cocks or their equivalent so that the proportions of the mixture can be roughly regulated by hand and so that the gas can be permanently shut off from the engine.

(d) The terms *mixing valve* and *proportioning valve* are used rather loosely to designate anything which has to do with the mixing of air with gas already measured out, or with the measuring and mixing of the constituents of the charge. In the strictest

sense a proportioning valve, and to a certain extent a mixing valve, precedes the inlet valve, measures the combustible part of the charge, and mixes it with the air. A gas valve under governor control, combined with surfaces, or passages, which will mix the gas with the air before or during passage through the inlet port, is properly a mixing or proportioning valve. One example of this sort of arrangement is shown in Fig. 294.

The small gas valve *a* is guided by the sleeve sliding on the stem of the inlet valve *b*. It is operated by separate linkage under governor control, so that the time, or extent, of its opening can be varied to suit the load. In operation, the inlet valve opens first, allowing fresh air to enter the cylinder and blow away hot burned gases. The gas valve *a* then opens, admitting gas, which, traveling downward, is thoroughly mixed with the air as it issues from the small holes shown. The valve *a* closes before the inlet valve *b*, so that the mixing chamber becomes filled with pure air before being shut off from the cylinder.

Such a device is commonly known as a *combined mixing and inlet valve*, although the gas valve is occasionally designated as a mixing valve or a proportioning valve.

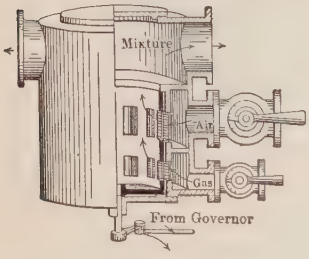


Fig. 295.

(e) The elements of another type of mixing valve are shown in Fig. 295. The inner cylinder is supposed to be under governor control, so that it can be rotated more or less as the load varies, thus changing the effective openings of the gas and air ports to suit the demand for power. By properly proportioning the gas and air ports, their areas may be made to change at the same rate under the action of the governor, thus giving throttling regulation; or the areas may be made to change differentially, giving quality governing or mixed quality and quantity regulation.

(f) Experience has shown that proportioning valves of the type shown in Fig. 295, and others using sliding surfaces, are perfectly satisfactory when used with such fuels as natural gas and illuminating gas. Producer gas and blast-furnace gases, however, carry impurities which quickly foul such sliding surfaces and impair the action of the valve. For such gases, mixing and

proportioning valves made without sliding surfaces, such as that shown in Fig. 294, must be used. Even the valve shown in this figure might give trouble because of deposits on the stem of the main valve, and a design eliminating this possibility would probably give better results.

214. Methods of Ignition. (a) In the early development of gas engines the charge was ignited by opening communication at the proper time between the compression space of the engine and a small chamber containing an **open flame.** This method was complicated mechanically, and had so many objectionable features that it did not survive.

(b) The methods at present used are:

(*a*) Hot-tube ignition;

(*b*) Spontaneous ignition by heat of compression (assisted, or not assisted, by the action of a hot chamber, such as a vaporizer or hot bulb);

(*c*) Electric ignition.

215. Hot-Tube Ignition. (a) A simple type of hot-tube ignition is shown schematically in Fig. 296. The tube *a*, generally made of metal, is closed at one end, while the other end opens into the cylinder. By moving the burner and chimney *b*, the hot zone, which is at about red heat, can be located anywhere along the tube.

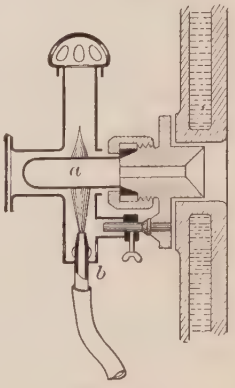


Fig. 296.

At the end of the exhaust stroke the hot tube, like the rest of the clearance space, is filled with burned gases at a pressure slightly above atmospheric. During the suction stroke these gases are partly expanded, and during the compression stroke they are compressed into the tube by the combustible mixture until the latter finally reaches the hot zone, where it is ignited. By moving the hot zone along the tube, the time in the compression stroke at which the mixture is ignited can be varied.

(b) By this method *ignition* is generally *certain*, but the *timing* is *untrustworthy* because of variations in the condition of the

tube or of the mixture. Hence, despite its simplicity and lack of moving parts, hot-tube ignition is not now very widely used. "Timing valves" have been used to close the cylinder end of the tube and to thus control ignition, but few have survived.

Hot-tube ignition involves a constant supply of gas to the burner, and this of course adds to the fuel consumption of the engine.

216. Spontaneous Ignition. In many engines using liquid fuels heavier than gasoline, ignition is produced by the temperature attained during compression. In the Diesel engine the compression pressure is so high that the resulting temperature alone causes ignition. In other engines, like the hot-bulb type (Fig. 289), ignition results from the combined action of compression and a hot vaporizing chamber.

This method of igniting has not proved applicable to the more volatile liquid fuels and to the gaseous fuels because of the difficulty of timing.

217. Electric Ignition. (a) The most satisfactory method of igniting is by an electric spark.

All electrical ignition systems in use, with few exceptions, fall under either "*make-and-break*" or "*jump-spark*" ignition. The less descriptive terms, "low-tension" ignition and "high-tension" ignition, are often used in place of these.

(b) In the **make-and-break ignition system,** two "*electrodes*" are brought together within the combustion space to "*make*," or close, the circuit, and are separated suddenly to "*break*" the circuit and produce a spark.

One arrangement of such a system is shown in Fig. 297, the "*igniter block*," or "*plug*," entering the combustion space through the center of the cylinder head. The "*stationary electrode*" is designated by *i* and the "*movable electrode*" by *j*. The wiring diagram is shown in Fig. 298. In this figure, *B* represents a battery or other low-voltage generator, *C* an "induction" or "*intensifier coil*," *E* the stationary electrode, which is insulated from the igniter block and engine frame, and *S* a stud or other convenient screw fastening on the engine. The movable electrode is in electrical contact with the igniter block and engine frame, as shown in Fig. 297.

(c) The operation is as follows: The cam *a*, Fig. 297, pushes

the rod *b* toward the igniter and the strike block *d*, engaging the flipper *e* on lever *f*, moves the latter toward the left. As *f* moves, it draws *g* after it by means of the one-turn spring shown. As *g* moves it rotates the movable electrode until the arm *j* inside of the cylinder is brought into contact with the stationary

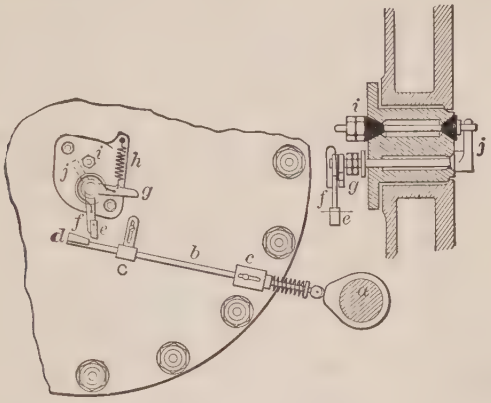


Fig. 297.

electrode *i*. The circuit is then made and current flows until the circuit is broken by the block *d* traveling past the edge of the flipper *e*. When this occurs, the spring *h* pulls the arm *j* out of contact with *i*, and the circuit is broken. The spark results from the action of the induction coil at the instant of breaking the circuit. The rapid change in the number of lines of force

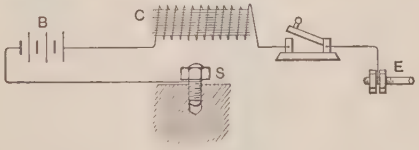


Fig. 298.

through the core causes sufficient self-induction to generate an electromotive force of such intensity as to bridge the gap between the separating electrodes.

(d) The timing of the spark is effected by moving guide *C* across the path of the bar *b* in Fig. 297, thus changing the time at which block *d* releases flipper *e*.

(e) The type of igniter just described is known as a "*hammer make-and-break igniter*" to distinguish it from another known

as a "*wipe-spark*" or "*wipe make-and-break igniter*," in which a movable electrode periodically wipes or slides across a stationary electrode. The wipe spark automatically cleans the contact surfaces within the cylinder, which is in a way advantageous, but it is not so extensively used as the hammer type.

(f) The make-and-break system has the advantages of being *electrically simple* and operating with *low e.m.f.*, so that short circuits are not so apt to occur as in the systems described in following sections. It is, however, *complicated mechanically*, and because of friction and inertia of parts is not generally used on engines operating at speeds above 500 to 600 r.p.m. The movable electrode is very apt to stick or to work loose, causing trouble because of no spark or because of loss of compression by leakage.

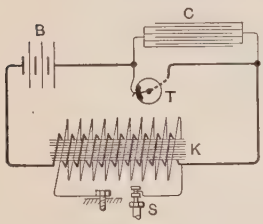


Fig. 299.

(g) In the **jump-spark system** there are within the cylinder two *fixed terminals*, with short intervening gap, across which a spark jumps when sufficient difference of potential has been developed. In its simplest form the apparatus has two circuits, as shown in Fig. 299, with heavy lines representing the "*low-tension circuit*" and the light lines the "*high-tension circuit*."

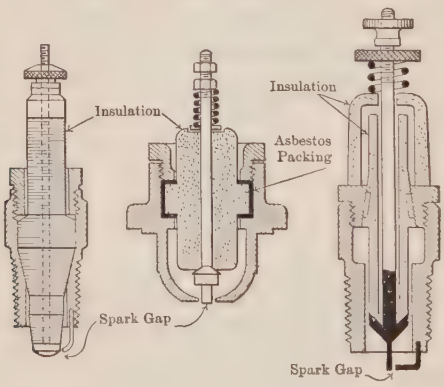


Fig. 300.

In the figure, *B* is the source of electromotive force, *T* is a rotating "timer," *C* a "condenser," *K* a "coil," and *S* a "spark plug," several examples of which are shown in Fig. 300.

(h) In *operation* the primary circuit is closed by the timer T and then suddenly opened, with the result that a spark jumps between the terminals of the plug. The action of the coil is as follows: When the primary circuit is closed by rotation of the timer, the magnetic field induces an electromotive force in the secondary circuit. This is not great enough, however, to cause a spark to pass between the plug terminals. But when the primary circuit is quickly broken, the sudden collapse of the magnetic field about the core of the coil induces for the instant in the secondary circuit a very high potential difference, which may be made sufficient to cause the passage of a spark, with resultant ignition.

The *function of the condenser*, which bridges the timer in the primary circuit, is to prevent sparking at the contact points of that apparatus. Such sparking would cause rapid deterioration of the contact surfaces and is therefore undesirable.

(i) A more common type of jump-spark apparatus uses a "*trembler coil*" instead of the plain induction coil shown in Fig. 299. This apparatus is so arranged that the trembler forms part of the primary circuit, and is in such position that it is attracted to the core of the coil when this is magnetized, and thus breaks the primary circuit. This in turn demagnetizes the core, hence the trembler flies back and makes the circuit once more; thus the core is again magnetized and attracts the trembler, breaks the circuit, and so on, as long as the timer is in position to close the primary circuit. This intermittent making and breaking of the primary circuit causes a succession of sparks at the spark plug in the secondary circuit, which action is generally supposed to insure more certain ignition. The great advantage achieved is really quick action and accurate timing, though these are often counterbalanced by considerable trouble with the trembler which may call for almost constant adjustment.

(j) Both of these high-tension or jump-spark systems are *easily timed* by shifting the phase relation of timer, or commutator, and engine crank, and they are particularly *satisfactory for high speed*. Recently there has been a tendency to adopt these systems for ordinary slow-speed stationary work; but as the spark does not seem to have the same igniting power as that of the make-and-break system, most applications have been limited to the more easily ignitable fuels like natural and illuminating

gas and gasoline. Few simple high-tension systems have yet been used with producer gas and "blast-furnace gas."

218. Internal-Combustion Engine Valve Gear. (a) The *slide valve*, so common in steam-engine practice, is never used in its simple form on internal-combustion engines for admission or exhaust. It is sometimes used for mixing purposes, as was indicated in Sect. 213. The high temperatures to which inlet and exhaust valves are subjected make lubrication difficult and cause warping of the valve and seat, and the high pressures make it difficult to keep the valve on its seat to prevent leakage. When the fuel used contains sulphur, which is not an uncommon occurrence, the valve and seat are often quickly pitted and corroded.

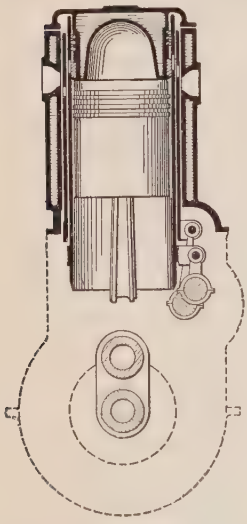


Fig. 301.

(b) Some highly specialized slide valves are, however, in use and give good satisfaction. The control of ports by the piston of the two-stroke-cycle engine is the most common example. Recently a number of "*sleeve motors*" have been designed for use on automobiles and seem to promise very satisfactory operation.

One example of this type is shown semi-diagrammatically in Fig. 301. The two sleeves, reciprocating vertically under the action of eccentrics or cranks on a side shaft, act in conjunction with the cylinder head and external cylinder to control admission and exhaust by means of the ports shown. The advantages of this type are rapid opening and closing of valves, long period of approximately maximum opening, and silent operation.

(c) The success of this type of valve has caused the appearance of a number of different varieties of slide-valve and piston-valve auto-engine designs. Few of these have been tested to any extent, and it is therefore too early to draw conclusions as to their ultimate success.

(d) With the exception of the cases cited above, the *poppet* or *mushroom valve* is in practically universal use for internal-combustion engines. It maintains its correct shape under

changing temperatures more perfectly than other types; it requires a minimum of contact surface between valve and seat; it opens inward and is therefore forced to its seat by the high pressures in such engines; it requires no lubrication; and it and its seat are easily kept comparatively true by grinding.

(e) In modern designs, inlet valves are practically never *water-cooled*, as the ingoing charge cools them sufficiently during each suction stroke. Exhaust valves, on the other hand, are practically always water-cooled when larger than five inches in diameter, and often in smaller sizes. This is deemed necessary because of the high temperature of the exhaust gases in which the valve is immersed during the entire exhaust period, but it should be noted in this connection that one European builder is obtaining satisfactory operation with simple uncooled cast iron exhaust valves in the largest sizes of horizontal engines.

(f) In some four-stroke-cycle engines the operating conditions of the exhaust valve have been improved by the use of "*auxiliary exhaust ports*." These are ports in the cylinder wall which are uncovered by the piston when near the end of its stroke. The first discharge of exhaust gases takes place through these ports, so that a smaller quantity of cooler gases is handled by the exhaust valve.

This construction necessitates the use of a larger cylinder for a given power than is required without the use of auxiliary ports, and it complicates the cylinder casting. It is practically never used on double-acting engines because of these reasons, and because of the additional fact that it would necessitate the use of an enormously long piston, similar to that shown in Fig. 280, thus materially increasing the weight of the reciprocating parts.

(g) Two types of inlet valve are in use, — the *automatic valve* and the *positively actuated valve*. The automatic valve is held to its seat by a weak spring, and is raised by the difference between atmospheric and suction pressures during the suction stroke. The positively actuated valve is opened mechanically and generally closed by spring pressure.

Automatic valves are uncertain in their action, opening only after a considerable pressure difference has been created, and then more or less slowly. After opening they do not remain wide open during the remainder of the suction stroke, but "chatter" more or less, thus materially decreasing the volu-

metric efficiency of the engine. For these reasons they are seldom used on the better types or on the larger engines.

Positively actuated valves, on the other hand, can be made to open at the time desired, can be given an amount of opening approximately equal to that theoretically required at each piston position, and can be made to close very nearly at the right time.

(**h**) The valves of internal-combustion engines are generally operated by means of *cams*, or *eccentrics*, on a side shaft, or auxiliary shaft, driven by gearing from the crank shaft. On the smaller engines cams are most often used, but on the larger engines the eccentrics seem to be preferred, particularly in this country. Closure practically always occurs by spring pressure, the valve being released by the opening mechanism.

The cam can be manufactured more cheaply than the eccentric, and when properly designed it is not very noisy in operation and wears slowly. In general, however, it is rather difficult to obtain as perfect valve operation with cams as it is with eccentrics unless linkage is introduced, which complicates the mechanism and increases the cost.

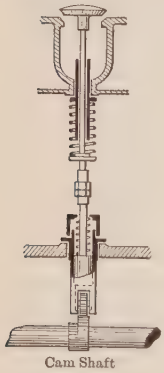


Fig. 302.

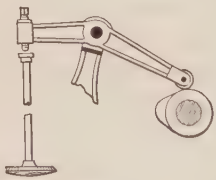


Fig. 303.

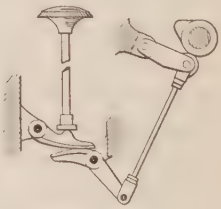


Fig. 304.

Cams may be used to operate the valves by direct contact with the valve stem (Fig. 302); or by contact with one end of a pivoted lever, the other end of which contacts with the valve stem (Fig. 303); or through *rolling, rocking, or floating levers*, one arrangement of which is shown in Fig. 304.

The eccentric always operates in conjunction with such levers as are shown in Fig. 304.

(i) The time (with reference to crank and piston positions) at which valves open and close varies widely with the location of the valve and with the type of engine. The exhaust valve universally opens early, generally when the piston is at about 0.9 stroke. It may close before the end of the return stroke, or on dead center, or it may remain open until after the suction stroke has started. The object of leaving it open after dead center has been passed is to take advantage of the inertia of the moving exhaust gases and thus get more perfect discharge. Where the valves, manifolds, and cylinders are so arranged that this can be done, it represents good practice. The inlet valve very commonly opens after the beginning of the suction stroke, though it is sometimes opened just before, or on dead center, in order to obtain a wider opening by the time suction actually starts. It is very generally closed after the end of the suction stroke in order to take advantage of the inertia of the moving column of gas, thus increasing the volumetric efficiency.

In general, the higher the speed of an engine the later may the valves close, and the greater may be the overlap of exhaust closure and inlet opening if the valves are widely separated.

(j) Because of the heavy springs necessary to close the valves of internal-combustion engines in the short time available, and because of the relatively great weight of the valves, the parts actuating the latter are generally very strong and heavy. This is particularly true of exhaust-valve gear. This valve must be opened against the combined action of high-pressure gas and a very powerful spring.

Many designers have attempted to reduce the size and wear of the actuating parts by building balanced exhaust valves. As a general rule these have not survived, probably because they simplify the external gear by complication of the inclosed part of the valve system.

Because of the great weight of the valves and actuating mechanisms in large engines and because of the great magnitude of the forces transmitted by these mechanisms, it is generally undesirable or even impossible to construct governors which can operate in any such direct manner as is common in the average steam engine. Governors could not be constructed powerful enough to operate directly unless made with such

heavy parts, and to transmit such great forces, that their sensitiveness would be considerably impaired.

In very large engines a differential governing device is now commonly used. In such cases the governor operates upon the equivalent of a small engine of some kind, which engine, in turn, supplies such power as is necessary for moving the valve gear. As an example, the governor might actuate a small pilot valve which by its motion admitted oil under pressure to one, or the other, end of a cylinder fitted with a piston suitably linked to the inlet- or mixing-valve gear. The motion of the piston in the proper direction and to the right extent, as controlled by the governor through the pilot valve, would then serve to give the required adjustment of the main valves.

In smaller engines it is customary to connect the governor to some light form of mixing valve, to a balanced or floating valve of some kind, or to a light link or equivalent which is easily moved and causes the necessary adjustment by the shifting of a fulcrum or the like in the main gear.

CHAPTER XXVI.

INTERNAL-COMBUSTION ENGINES (*cont.*).

Efficiency, Performance, and Power.

219. Efficiencies of Otto Four-Stroke Cycle Engines. (a) Not only does the thermal efficiency of the Otto cycle engine theoretically vary with the ratio of compression, increasing as the final volume is decreased with respect to the initial volume, but real engines also show a similar gain. The rapid improvement in the efficiency of this type of engine during the past twenty years has been largely due to this increase in compression pressure. It is well shown by the following table:*

TABLE XII. — EFFICIENCIES OF OTTO FOUR-STROKE CYCLE ENGINES.

No.	Year.	Type of Engine.	Cylinder Size.	Indicated Thermal Efficiencies.	Brake Thermal Efficiencies.	Mechanical Efficiencies.
			Inches.	Per cent.	Per cent.	Per cent.
1	1882	Deutz	6.75 × 13.7	16	14	87.6
2	1888	Crossley	9.5 × 18	22	18.9	86.1
3	1898	National	10 × 18	28.7	25	87.0
4	1908	Crossley	11.5 × 21	36.8	32.2	87.5

(b) It should not be assumed, however, that by an indefinite increase of compression pressure the thermal efficiency of the real engine can be raised without limit. For even if the tendency of the fuel to preignition could be overcome, calculations based upon actual performances show that with the Otto type of engine the maximum practical thermal efficiency would probably be attained with a compression pressure of from 250 pounds to 300 pounds per square inch.

Blast-furnace gas engines operating with compression pressure as high as 200 pounds have given thermal efficiencies on the brake of 32 to 34 per cent. But the tendency with this fuel is

* The Gas, Petrol and Oil Engine, D. Clerk, page 243.

now toward the use of compression pressures in the neighborhood of 160 to 180 pounds because of the mechanical difficulties encountered with the higher pressures; and in this case a little under 30 per cent is extremely good thermal efficiency on the brake for modern engines, while the average operating value for good standard American types of stationary engines is about 25 to 27 per cent at rated load, and of course decreases with reduction in the load.

(c) Besides the compression ratio, the thermal efficiency in general can also be increased by

(1) Mixing the incoming charge more perfectly;

(2) Producing fairly rapid and complete combustion at the compression end of the stroke (note, however, that too rapid combustion is not desirable);

(3) Preventing loss of heat from the charge to surrounding metal during combustion and expansion.

Many modern engines have elaborate mixing valves which cause thorough intermixing of gas and air before, or just at the time of, entering the cylinder.

In high-efficiency engines the combustion space is made as nearly as possible spherical, hemispherical, or in the form of a short cylinder; and all pockets leading out of this space are avoided as far as possible. This results in less surface for the volume inclosed, and thus reduces heat loss to the metal and makes the combustion more rapid and complete for a similar reason.

In pockets connecting with the combustion space the gases often burn long after combustion of the main part of the charge is complete. This can be prevented by placing the igniter in the pocket, and igniting the gas there first, in which case the rapid increase of temperature will cause a sudden pressure rise, blowing some of the burning gas into the main charge, thus causing very complete inflammation.

Large engines generally have slightly higher thermal efficiencies than small engines of the same type and proportions, because large cylinders have less wall surface per unit of volume inclosed than have small cylinders of the same proportions. This, however, may be counteracted by difficulty of mixing the charge in the larger cylinder and difficulty in effecting rapid and complete combustion.

When large cylinder diameters are used, two or more igniters

at different points are often operated simultaneously in each combustion space in order to reduce the distance through which inflammation must progress from each igniter.

Piston speeds of high-efficiency engines are carried as high as is mechanically feasible in order to reduce the time of contact between hot gases and walls.

(**d**) The values of all the different efficiencies enumerated in Sect. 105 will vary considerably with the conditions, fuel, mixture, type of engine, etc.; but for the purpose of giving a general idea of the order of these values a certain type and set of conditions will be assumed.

The engine is supposed to operate with "producer gas" as fuel and (in the ideal case for drawing the air card) to have a suction pressure equal to atmospheric, a pressure of 150 pounds per square inch absolute at the end of compression, a temperature at the end of suction stroke equal to 520° F. abs., a temperature at the end of compression of 1000° F. abs., and a temperature at the end of combustion of about 6500° F. abs. These figures are obtained by neglecting all losses in the real engine and by considering the specific heats constant.

(**e**) The *thermodynamic or Carnot efficiency* is then

$$Efc = \frac{T_1 - T_2}{T_1} = \frac{6500 - 520}{6500} = 92 \text{ per cent.}$$

(**f**) The *cycle efficiency* for this Otto cycle is from Eq. (80),

$$CEf = 1 - \frac{T_d}{T_a} = 1 - \frac{520}{1000} = 48 \text{ per cent.}$$

Then in Fig. 305, drawn to scale for the assumed engine, the distance AB is 48 per cent of AC.

Thus the Otto cycle upon which this engine is to operate is less efficient than reversible cycles, and the real engine is initially handicapped to that extent.

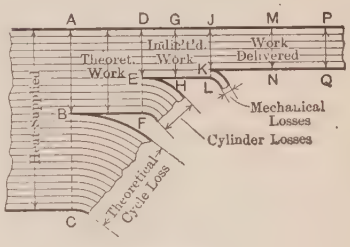


Fig. 305.

(**g**) The *relative efficiency* is

$$REf = \frac{CEf}{Efc} = \frac{48}{92} = 52.2 \text{ per cent.}$$

This shows that the real Otto engine, if absolutely perfect, could only make available a little more than half the mechanical energy obtainable with the ideal Carnot engine.

(h) The *indicated efficiency* measures the amount by which the cylinder of the real engine falls short of developing the 48 per cent of the supplied energy.

The weight (W_1) of mixture that this engine would probably use is about 9 to 10 pounds per i.h.p.-hour, and the heat ΔQ supplied by each pound of mixture is about 940 B.t.u. Then the theoretical Otto engine would make available $940 \times 0.48 = 451.2$ B.t.u. per pound as ΔE. One horse power is equivalent to 2545 B.t.u. per hour, and the heat theoretically available for doing work is (9 or 10) $\times$ 451.2 B.t.u.; hence

$$IEf = \frac{2545}{W_1 \Delta E} = \frac{2545}{(9 \text{ to } 10) \times 451.2} = 62.6 \text{ to } 56.4 \text{ per cent.}$$

That is, the area of the upper loop of the real indicator card divided by the area of the ideal air card would give a value between 62.6 per cent and 56.4 per cent. This measures the proportion of the maximum energy of this cycle that is made available by the real engine. In Fig. 305, DE should be 62.6 per cent to 56.4 per cent of DF.

(i) The *thermal efficiency on the i.h.p.* is easily determined to be

$$TIEf = \frac{2545}{W_1 \Delta Q} = \frac{2545}{(9 \text{ to } 10) \times 940} = 30 \text{ to } 27 \text{ per cent,}$$

which shows that the real engine actually converts into mechanical energy from 30 to 27 per cent of all the heat supplied it. Some of this is, however, lost in fluid and mechanical friction, and the amount of such loss is measured by the mechanical efficiency.

The $TIEf$ is the ratio of GH to AC in Fig. 305.

(j) The *mechanical efficiency*, MEf, of an engine of this kind would probably be about 85 per cent, thus the d.h.p. would be about 85 per cent of the i.h.p. In Fig. 305, JK is therefore 85 per cent of JL.

(k) The *thermal efficiency on the d.h.p.* is from Eq. (220)

$$TDEf = TIEf \times MEf = (27 \text{ to } 30) \times 0.85 = 22.9 \text{ to } 25.5 \text{ per cent,}$$

showing that the engine actually turns into useful, available power about one quarter of all the heat energy supplied it. In Fig. 305 the $TDEf$ is given by the ratio of MN to AC.

(1) The *over-all efficiency* would be by Eq. (221)

$OEf = IEf \times MEf = (56.4 \text{ to } 62.7) \times 0.85 = 47.9 \text{ to } 53.3$ per cent,

showing that the real engine losses (cylinder, fluid friction, and mechanical friction) consume about one-half the power which the ideal engine with the same cycle would make available. In Fig. 305 the OEf is the ratio of MN to AB.

220. Efficiencies of other Commercial Engines. (a) **Two-stroke-cycle Otto Engines,** because of greater cyclinder and friction losses, generally have over-all efficiencies of from 0.7 to 0.8 of those of corresponding four-stroke engines. The indicated efficiency and mechanical efficiency may both be lower than in four-stroke engines, or the indicated efficiency may be lower while the mechanical efficiency is higher because of the absence of valves and such.

(b) The thermal efficiency of the **Diesel oil engine** is generally higher than that of engines working on the Otto cycle. This is due to the higher compression pressure which can be carried in these engines (500 pounds per square inch or more), and to the fact that the combustion conditions are also probably somewhat better.

Average thermal efficiencies on the brake with Diesel engines are about 30 per cent, and sometimes run as high as 35 per cent.

221. Heat Balance for Gas Engines. (a) In reporting an engine test, it is customary to account for all heat supplied. The statement of this account is called the "heat balance." There are only five possible destinations for heat supplied to a gas engine. They are:

(1) Useful mechanical energy;
(2) Loss to jacket;
(3) Heat carried away in the exhaust gases;
(4) Loss due to incomplete combustion;
(5) Radiation, which includes energy converted into heat by friction.

(b) The useful mechanical work has already been shown to equal from 15 to 30 per cent of the heat supplied.

(c) The relative amount of heat lost to the water, or air, jacketing the cylinder varies in different engines, and in the

same engine under different conditions. The loss to jacket is between 25 and 50 per cent, with an average from 30 to 35 per cent. In the case of air jacketing, it is not generally possible to distinguish between jacket and radiation losses.

(d) The loss due to heat carried away by the exhaust gases, owing to their high temperature, generally falls between 25 and 40 per cent, increasing as the jacket loss decreases, and *vice versa*.

(e) Combustion is almost always incomplete to a small extent and may at times be imperfect enough to account for a considerable proportion of the heat available. This loss should not be greater than 1 to 2 per cent of the total heat, and is often much less.

(f) Radiation loss is supposed to include all heat radiated from the outer surfaces of the engine, and in the heat balance it would include all energy converted into heat by friction and subsequently lost by radiation and conduction. It is generally found by subtracting the sum of the other four quantities of heat in per cent from 100; and when this method is used this difference includes all errors of the other results. When calculated in this way it may have a value of from 10 to 20 per cent, with an average of about 15 per cent.

(g) Another heat-balance method puts under (1) the energy represented by the upper loop of the diagram, instead of the mechanical energy delivered. Then the energy loss in gas and engine friction is already included under (1) and does not appear as radiation loss under (5). The latter value is then reduced to about 5 to 8 per cent of the total heat supplied.

(h) The total heat supplied the engine may be taken as either the higher or lower heat value of the gas (see Chapter XXVIII). Obviously the use of the lower value results in a higher efficiency for the engine, and is therefore favored by gas-engine builders. In America the lower value is universally used, although in some countries of Europe the higher value is sometimes adopted.

(i) It is important to note that the thermal efficiencies of steam and internal-combustion engines are not strictly comparable unless the amounts of heat available are measured in a truly comparable way. This is usually not the case, for the following reasons: The heat supplied a steam engine is generally figured as that in the steam above some datum, such as 32° F., or feed-water temperature, or exhaust temperature and is not in

terms of the fuel used or its cost. On the other hand, the heat supplied an internal-combustion engine is based upon a calorimetric determination of the fuel, with certain corrections in case the lower calorific value is sought. This amounts to figuring the heat supplied above a datum equal to the existing atmospheric temperature for all the constituents of the exhaust gas excepting the water formed by the combustion of hydrogen. The heat value of this combustible is figured above a datum which often corresponds roughly to 212° F. (See Chapter XXVIII for further discussion.)

The datum used is thus arbitrarily chosen for convenience in each case, and the results are not strictly comparable. It might seem that, since the steam engine is given credit for the heat of the liquid in the exhaust steam, or for that part of it above feed-water temperature, some sort of similar device might be adopted in the case of the internal-combustion engine. This is incorrect, however, because the exhaust of the latter engine is absolutely useless so far as the engine is concerned. Part of the heat carried may be abstracted by generating steam, heating water, or in a number of other ways; but this should not affect the figure for heat consumption of the engine, although it is properly taken account of in determining the efficiency of the plant as a whole.

(j) The only true comparison of heat expenditure is between heat-power plants as a whole and not between engines only. If the fuel is the same in both cases, the ratio of the amounts of fuel per d.h.p. may be used; otherwise relative economy is shown by the ratio of the costs of the respective amounts of fuel consumed per d.h.p.-hour.

The true comparison for economic purposes should include not only the fuel cost, but expenditure for labor, lubricants, supplies, repairs, interest, depreciation, insurance, and all other costs involved in power generation; and only on such a basis are two systems truly comparable.

222. Performance of Internal-Combustion Engines. (a) There are so many different kinds of internal-combustion engines that it is difficult to make broad statements to fit all cases. The following must, therefore, be regarded as very general, and applicable only to the average lines of engines.

(b) American engines built to run on **natural gas** are generally guaranteed to deliver a brake horse power on from 10 to 11 cubic feet of gas at rated load. This gas is commonly assumed to have a calorific value (lower) about 1000 B.t.u. per cubic foot; so this guarantee is from 10,000 to 11,000 B.t.u. per horse-power hour at rated load, corresponding to thermal efficiencies of from 23 to 25.5 per cent on the d.h.p. Many engines at present in operation give better results than these by several per cent at rated loads; and the efficiencies are still better at loads from 10 to 15% greater than the normal.

At three-quarter load they are generally guaranteed at 11,000 to 13,000 B.t.u.; at half-load, 13,000 to 15,000; and at one-quarter load 20,000 to 23,000 B.t.u.

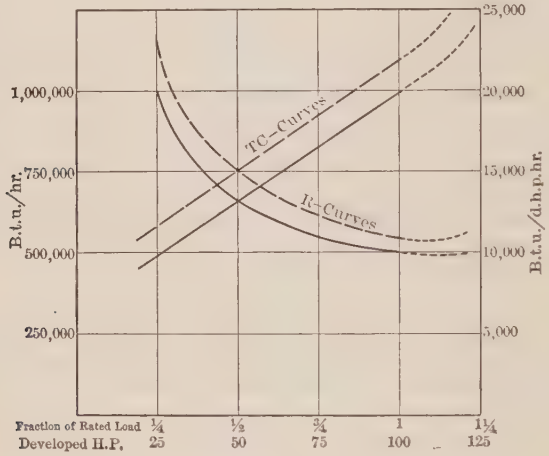


Fig. 306.

The curves of the *total consumption* and *rate* per d.h.p.-hour for average 100 horse-power natural-gas engines are given in Fig. 306. In each case the two curves correspond to the limits above given. The exact shape of these curves will, of course, depend upon the type of engine, method of governing, etc., but those given may be taken as representing average practice.

It is convenient to remember that practically all internal-combustion engines (with the possible exception of some oil engines) will require about twice as many thermal units per horse-power hour at one-quarter load as at the rated load.

(c) Engines intended to operate on **illuminating gas** are generally guaranteed with lower efficiencies than natural-gas engines. The B.t.u. per d.h.p.-hour is usually from 12,000 to 13,000 B.t.u. at full load. The poorer performance is principally due to the fact that these engines are, as a rule, not so carefully designed, as they are not built in large sizes or in great numbers because of the high cost of this gas. Some of the highest thermal efficiencies on record have, however, been obtained with engines using illuminating gas.

(d) **Producer-gas** engines are generally guaranteed on a basis of coal used per horse-power hour rather than cubic feet of gas or B.t.u. The average figure is 1 to 1.1 pounds of coal per horse-power-hour at rated load, and most producer-gas installations of good design can be counted on to produce a d.h.p.-hour on less than 1.2 pounds if operated continuously at full load. Under accurate test many of them have developed a brake horse-power hour on 0.8 to 0.9 of a pound of coal.

To give an idea of the meaning of these figures, it is sufficient to state that a consumption of only 0.8 pound of coal per d.h.p.-hour corresponds to a thermal efficiency on the brake for the engine alone of about 31 per cent; while 1 pound corresponds to about 25 per cent.

(e) **Gasoline engines** (stationary) are generally guaranteed to deliver a d.h.p.-hour on one pint of gasoline, at rated load. This corresponds to a heat consumption of about 14,000 B.t.u. per d.h.p.-hour, or a thermal efficiency of about 18 per cent. As a matter of fact, all of the better types are capable of delivering a d.h.p.-hour on about two-thirds of the guaranteed quantity, when everything is in perfect adjustment.

Between rated oad and maximum load the efficiency will first increase and then due to the use of rich mixture will slowly decrease.

At about one quarter load the consumption per d.h.p. will be about twice that at full load.

(f) **Alcohol engines** are not as yet a commercial product in this country, and very few figures are available from practice. Tests show that such engines can safely be guaranteed on the same or a smaller volume consumption than gasoline. It is safe to assume a thermal efficiency of 25 per cent on the brake with these engines, and figures as high as 32 per cent and more have

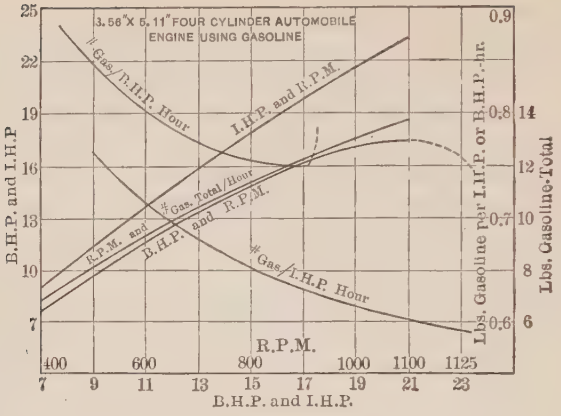

3.56"X 5.11" FOUR CYLINDER AUTOMOBILE ENGINE USING GASOLINE
Gas/B.H.P. Hour
I.H.P. and R.P.M.
Gas. Total/Hour
R.P.M. and B.H.P.
B.H.P. and R.P.M.
Gas/I.H.P. Hour
B.H.P. and I.H.P.
Lbs. Gasoline per I.H.P. or B.H.P.-hr.
Lbs. Gasoline-Total
R.P.M.
400 600 800 1000 1100 1125
7 9 11 13 15 17 19 21 23
B.H.P. and I.H.P.

Fig. 307.

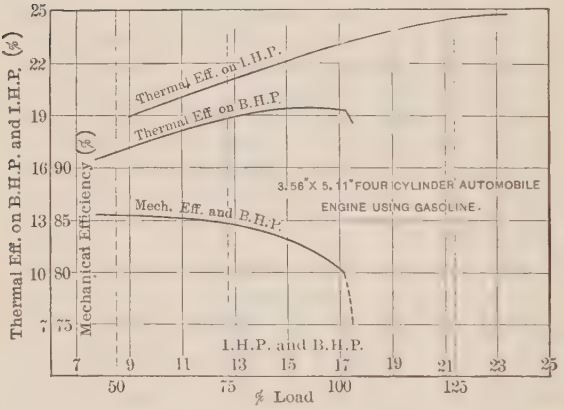

Thermal Eff. on B.H.P. and I.H.P. (%)
Thermal Eff. on I.H.P.
Thermal Eff. on B.H.P.
Mechanical Efficiency (%)
Mech. Eff. and B.H.P.
3.56"X 5.11" FOUR CYLINDER AUTOMOBILE ENGINE USING GASOLINE.
I.H.P. and B.H.P.
% Load
50 75 100 125

Fig. 308.

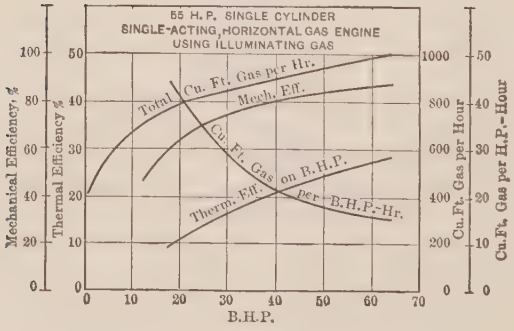


Fig. 309.

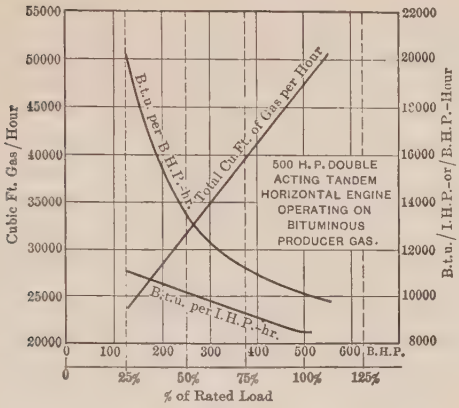


Fig. 310.

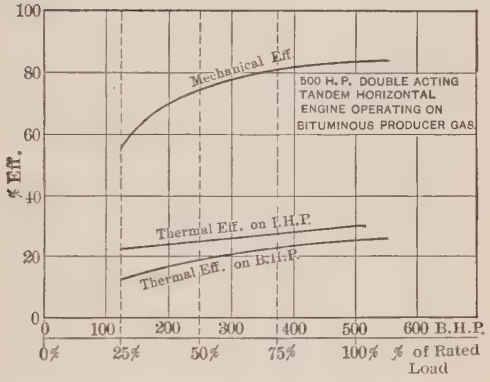


Fig. 311.

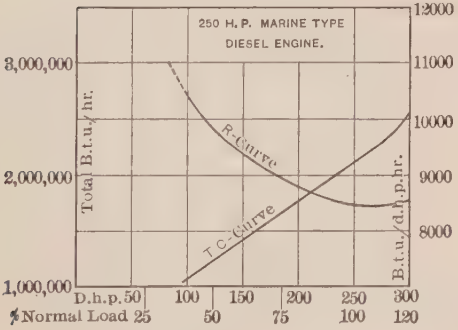


Fig. 312.

been obtained. This high efficiency is largely due to the high compression pressure that can be used with this fuel.

(g) **Oil engines** (kerosene, distillate, and crude) differ widely in fuel consumption, but the newer and better American types are capable of producing a d.h.p.-hour on from 0.7 to as low as 0.5 of a pound of oil. These figures correspond roughly to thermal efficiencies of from 18 to 28 per cent on the d.h.p.

(h) The curves given in Figs. 307 to 312 inclusive show results of tests of several different types of engines with different fuels. They illustrate in a general way how the various efficiencies of commercial importance vary with such things as load, size of engine, kind of fuel, etc.

CHAPTER XXVII.

FUELS.

223. Fuels. (a) In the discussion of ideal engines in preceding chapters, a hot body was assumed to be available and it was imagined to be so constituted that it could deliver heat at any time and in any desired quantity, with no change in its own temperature. No such hot body is really available, and in practice supplies of heat are obtained by burning "fuel."

(b) In the broadest sense *fuel is any material which can be made to combine with other material in such a way as to liberate heat.* In the commercial sense, however, fuel is any material the greater part of which can be made to combine with *oxygen*, usually from the air, so as to liberate heat, and which is purchasable at such a price that its use will yield a profit.

(c) Fuels may be solid, liquid or gaseous. The principal *Natural Fuels* are Coal, Wood, Petroleum Oil or Crude Oil, and Natural Gas. The principal *Prepared Fuels* are Coke, Briquets made from coal, Charcoal, Distillation Products of Petroleum, Artificial Gas made from solid or liquid fuel, Hydrogen Gas and Acetylene Gas made from noncombustibles, and Alcohol. There are also certain kinds of municipal refuse and manufacturing wastes which have fuel value.

224. Geology of Coal. (a) **Formation.** Beds of coal in the different stages of formation are scattered over the earth's surface. Geologists believe that coal results from collections of vegetable matter, deposited in swampy places or under water, which are subsequently covered by silt and other material and during geological ages are gradually changed in physical and chemical composition until they finally become coal.

(b) Vegetable matter may here be assumed to consist of carbon, hydrogen and oxygen combined in definite proportions, together with certain incombustible inorganic salts in the cell structure. This vegetable matter when under water changes

very gradually, losing some of its material in the form of gas (usually methane or marsh gas, CH_4) and as water. These transformations continue after the deposit has been deeply covered with earth, and eventually only the carbon and the inert salts remain. The extent of these changes is principally dependent on time, measured in geological ages, on temperature, and on the pressure, depth and porosity of the overlying material.

The *combustible* part of coal consists principally of *volatile matter* (which is released upon heating to a high temperature in

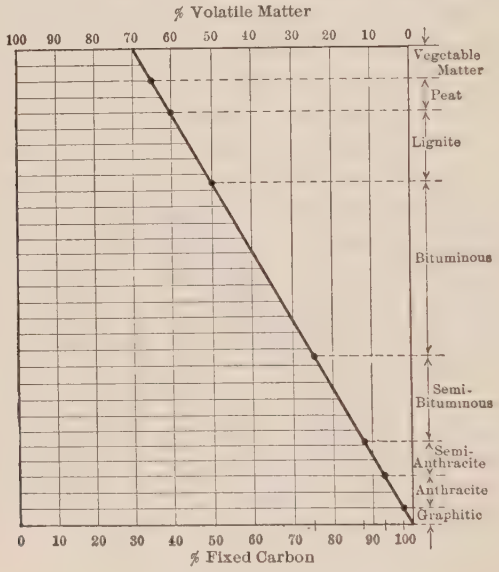


Fig. 313.

a closed crucible) and of *fixed carbon* which remains after such treatment. As the formation of coal progresses the percentage of volatile matter and moisture decreases with corresponding increase of fixed carbon.

(c) **Classification.** In the early stages of transformation the material is called (1) *Peat* or Turf. Later, with increased pressure of overlying material resulting in greatly reduced volume, it becomes (2) brown or black *Lignite*. Later still, after additional physical and chemical changes, the material becomes (3) Soft Coal or *Bituminous Coal*. Subsequently, it becomes

successively (4) *Semibituminous*, then (5) *Semianthracite*, (6) *Anthracite*, and finally (7) *Graphitic Coal*. The last is practically pure carbon. These are the seven groups into which coals are generally classified.*

(d) Fig. 313 shows in a *very general* way the relation of fixed carbon to volatile matter during the transformation of vegetable matter into coal.

The horizontal width of the diagram represents the sum of fixed carbon and volatile matter. The inclined line divides the horizontals into parts which represent fixed carbon (at the left) and volatile matter (at the right). Percentages may be read from the scales.

The progress of the transformation is shown by the classification at the right of the diagram. This grouping would seem to indicate well-defined divisions between adjacent classes; but in reality the groups blend into each other. The diagram is simply for illustration and should not be used otherwise.

(e) There is as yet no really satisfactory basis for the classification of coal. Formerly the classification was according to the percentage of fixed carbon in the dry *combustible*, as given in Table XIII. This, however, is not very satisfactory for coals high in volatile matter.

TABLE XIII. — OLD CLASSIFICATION OF COALS.

Kind of Coal.	Fixed Carbon.	Volatile Matter.
	Per cent.	Per cent.
Anthracite	97.0 to 92.5	3.0 to 7.5
Semianthracite	92.5 to 87.5	7.5 to 12.5
Semibituminous	87.5 to 75.0	12.5 to 25.0
Bituminous, Eastern	75.0 to 60.0	25.0 to 40.0
Bituminous, Western	65.0 to 50.0	35.0 to 50.0
Lignite	Under 50.0	Over 50.0

(f) A recently proposed classification, based on the ratio of volatile carbon to total carbon and known as Parr's Classification,† appears to be more satisfactory. Omitting the subgroups under bituminous coals and lignites, this classification is given in Table XIV.

* One other group falling between (2) and (3) above and known as "*subbituminous*" is sometimes recognized.

† Bull. No. 3 Illinois State Geol. Survey.

TABLE XIV.— PARR'S CLASSIFICATION OF COALS. (*Abbrev.*)

Kind of Coal.	$\frac{\text{Volatile Carbon}}{\text{Total Carbon}}$.	Inert Volatile.
	Per cent.	Per cent.
Anthracite	Below 4	.....
Semianthracite	Between 4 and 8	.....
Semibituminous	Between 10 and 15	.....
Bituminous	22 to 44	5 to 16
Lignite	27 and up	16 to 30

(g) Coal Fields in the United States.* The main deposits in this country are shown in a very general way in Fig. 314, in which the average character of each deposit is indicated by the

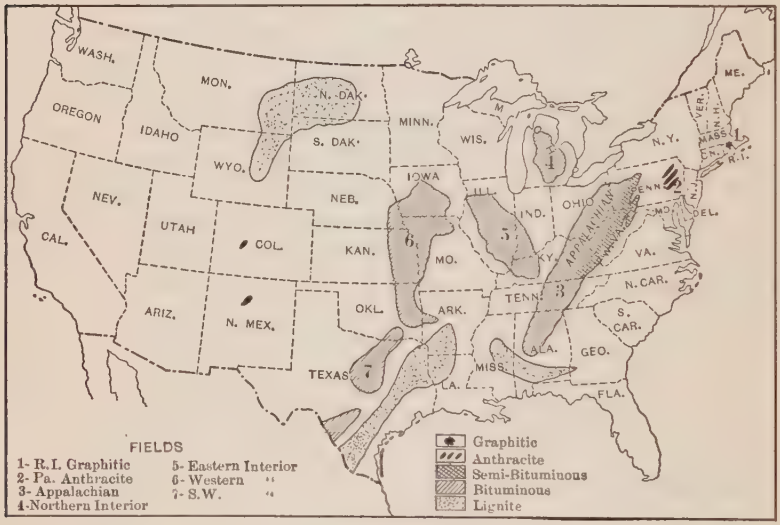


Fig. 314.

kind of hatching. In Rhode Island there is a little graphitic coal. Most of the anthracite is found in beds of less than 500 square miles area located in eastern Pennsylvania. The principal deposit of semibituminous coal is about three hundred miles long by twenty wide and lies along the eastern edge of the Northern Appalachian Field. The bituminous coals extend from this deposit westward. Starting with the graphitic coal in

* See Coal Fields in the U. S. by C. W. Hayes, U. S. Geological Survey, and Kent's "Steam Boiler Economy."

Rhode Island, broadly speaking, the farther west a coal is located the less advanced it is in the process of transformation. It is important to note, however, that there are many exceptions to these very general statements, for there are numerous other small fields, not shown, scattered over the country. For instance, a little anthracite coal is found in Colorado and in New Mexico, and some semibituminous in Arkansas.

225. Composition of Coal. (a) Coals consist principally of the elements Carbon, Hydrogen, Sulphur, Oxygen, and Nitrogen, together with moisture and ash. The elements named, particularly Carbon, Hydrogen and Oxygen, seem to be combined in various ways in the solid coal, though little is known of the formulas of the compounds in which they exist. The *ash* contains the inert salts of the original vegetable matter, together with silt and similar impurities acquired after deposition and submersion.

(b) "*Moisture*" is arbitrarily defined as the material lost when a finely powdered sample of the coal is maintained from half an hour, to an hour, at a temperature of about 220° Fahr.; or, more exactly, as the maximum loss which can be made to occur at this temperature. The material driven off in this way is not necessarily all moisture, for, with some coals, part of the more volatile combustible material may distil off. Moreover, all the water content may not be driven off by maintaining the material at this temperature. The definition is, therefore, only an arbitrary one, but it seems to be the best that can be devised.

(c) "*Dry Coal*" is coal from which the moisture has been driven by heating, as above described.

(d) "*Volatile Matter*" (or "*volatile*") is the name given to all material driven off when "dry coal" is maintained at a very high temperature (between a "red" and "white heat") in a covered crucible (out of contact with air) until there is no further loss of weight. This definition is again purely an arbitrary one, but it is useful in that it gives a measure of the material which will be similarly given off in a furnace or in a coke oven.

(e) "*Fixed Carbon*" is defined as the portion remaining after subtracting the ash from the material left in a crucible after driving off the volatile matter.

(f) "*Combustible*" is the term used to designate the part of the coal other than moisture and ash. It is, therefore, the sum of

fixed carbon and " volatile," as above defined. It is composed principally of carbon and hydrocarbons but it is important to note that it also contains noncombustible matter such as Nitrogen and Oxygen and hence the term is a misnomer. When the coal contains sulphur a large part of this is also found in the so-called combustible.

226. Coal Analyses. (a) Two types of analysis are in common use — one gives what is known as an " Ultimate Analysis," the other a " Proximate Analysis."

(b) In an *ultimate analysis* of so-called "dry *combustible*" the percentages of Carbon, Hydrogen, Oxygen, Nitrogen, and Sulphur are determined. The ultimate analysis of "dry *coal*" also includes the percentage of ash, and in some cases a chemical analysis of the ash is also made. Ultimate analyses are seldom made by engineers, being more often obtained from chemical laboratories. Table XV gives in a general way the approximate ranges of the ultimate analyses of the combustible in different kinds of coals. More accurate tables of analyses of coals from different localities can be found in text books on fuels and on boilers, in engineer's " pocket books " and in reports and publications of the geological surveys of the United States and of various states. In consulting such references it is necessary to bear in mind that ultimate analyses are sometimes incorrectly made on the basis of coal " as received," i.e., on wet coal. In such cases the percentages of *H* and *O* include the hydrogen and oxygen of the moisture.

TABLE XV. — ULTIMATE ANALYSES OF COALS.

	Per Lb. of Dry Combustible.				
	C	H	O	N	S
Anthracite	92–98	1–3.5	2–3	1	0–1.5
Semianthracite	90.5	5	4.5	.....	.....
Semibituminous	87.3	4.5–5.5	3–4.8	0.9–1.8	0.6–1.3
Bituminous	75–83	5–6.8	4–11	1–2	0.4–3
Lignite	70–78	5	10–15	2	1–3
Peat	61	6	33	.....	.....

(c) The *proximate analysis* divides the fuel roughly into the several parts which have already been described in Section 225,

as Moisture, Volatile Matter, Fixed Carbon, and Ash. While this analysis is less exhaustive than an ultimate analysis, it has two marked advantages over the latter: (1) It is easily made by the engineer and involves the use of very simple apparatus; and (2) it indicates, in a general way, the behavior which may be expected of the coal during utilization as fuel.

(d) Proximate analyses are given both on the basis of "dry coal" and coal "as received." For purposes of comparison with other fuels, the dry-coal basis is the better because the conditions of storage and transportation may materially change the moisture content. It would be obviously unfair, for instance, to charge against a coal, in comparison with others, the fact that it had been rained on, or had been stored under water. On the other hand, when coal is being purchased by weight, it is as obviously unfair to pay for water at the price of coal, and therefore for this and similar purposes analyses should be on a basis of coal "as received," or else, in addition to the proximate analysis of dry coal, there should be a statement of moisture content.

(e) Table XIII gives the approximate range of percentages of fixed carbon and volatile in the *combustible* of the different kinds of coal. The proximate analyses of "dry coals" may be obtained by introducing average ash contents and altering the percentages accordingly. Proximate analyses of coal from different localities may be found in the books and reports to which reference has already been made.

(f) It will be shown in later chapters that such data as are given by the ultimate analysis can be used for calculating the calorific value, and that they are also needed for computing losses occurring in furnaces and boilers. For these reasons ultimate analyses are often desired, even though their actual determination is outside the engineer's field. It has been shown by Professor L. S. Marks* that, in the case of most coals occurring in the United States, the ultimate analysis can be approximated from the proximate analysis with sufficient accuracy for determining the distribution of boiler and furnace losses and for general engineering work. The results which Professor Marks gave by curves have been put into the form of equations by Professor

* *Power*, vol. 29, p. 928, Dec., 1908.

H. Diederichs. Only the principal ones of these equations will be given here.*

Letting V represent the weight-percentage of volatile matter in the combustible, then the approximate weight-percentages of hydrogen (H), of volatile carbon (C), and of nitrogen (N) are respectively

$$H = V\left(\frac{7.35}{V + 10} - 0.013\right). \quad (329)$$

$$\left.\begin{aligned} C &= 0.02\ V^2 \\ &= 0.9\ (V - 10) \end{aligned}\right\} \text{for anthracite and semianthracite.} \quad (330)$$

or

$$C = 0.9\ (V - 14) \text{ for bituminous and semibituminous} \quad (331)$$

$$C = 0.9\ (V - 18) \text{ for lignites.} \quad (332)$$

$$N = 0.07\ V \text{ for anthracite and semianthracite.} \quad (333)$$

$$N = 2.10 - 0.012\ V \text{ for bituminous and lignite.} \quad (334)$$

The occurrence of oxygen and sulphur is apparently more or less accidental in character, showing no uniformity, and is not expressible by equations. The greater part of all the sulphur and some of the oxygen will appear in the proximate analysis as volatile, and will therefore be accounted for as hydrogen and carbon in the use of these equations.

227. Fuel Values of Coals. (a) The methods of determining the fuel values of combustible materials will be discussed in detail in the next chapter; there are a few considerations, however, which it is necessary to mention briefly at this point. It is customary to state the calorific value of a material in terms of the B.t.u. made available by burning one pound. When a material containing uncombined hydrogen is burned, this hydrogen unites with the oxygen and forms superheated water vapor. If this vapor passes off without surrendering its heat, the calorific value of the fuel is less than if that heat is made available. Hence the terms *lower heat value* and *higher heat value* are used to distinguish between the two conditions of combustion.

(b) As will be explained in Chapter XXVIII, the *Calorific Value* of a coal can be very roughly determined from the ultimate analysis by the use of *Dulong's Formulas*. These are given

* For more complete explanations, percentages of accuracy, etc., see Carpenter and Diederichs' "Experimental Engineering," p. 507, and the original article in *Power* referred to in the preceding footnote.

in Section 243 as Eqs. (376) and (377) and are stated as follows:

$$\text{Higher B.t.u.} = 14{,}600\,C + 62{,}000\,(H - O/8) + 4000\,S.$$
$$\text{Lower B.t.u.} = 14{,}600\,C + 52{,}000\,(H - O/8) + 4000\,S.$$

If for C, H, O, and S are substituted the weights of these elements per pound of *combustible*, the results will be B.t.u. per pound of combustible. If weights per pound of *dry coal* are used, the result will, of course, be B.t.u. per pound of dry coal.

As will be explained more in detail in the next chapter, Dulong's formulas are only approximate because they assume all the oxygen originally present in the fuel to be combined with

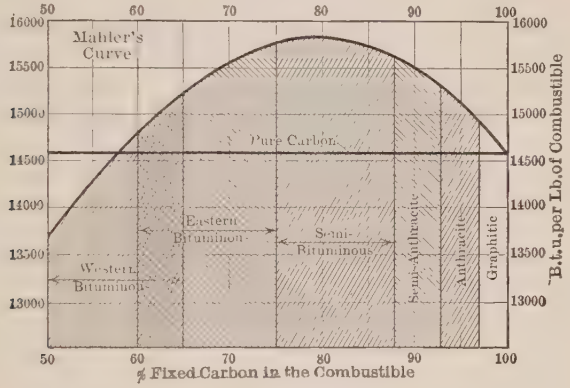


Fig. 315.

hydrogen, and because they take no account of the disappearance of heat accompanying the dissociation of hydrocarbons and similar obscure phenomena occurring during combustion. For accurate determination of the calorific value, some form of *fuel calorimeter* should be used. These calorimeters will be discussed in Section 244.

(c) Various *empirical formulas* have been proposed for giving the heat value per pound of fuel in terms of the proximate analysis. These usually contain "constants," which are given in tables, and which vary with the locality of the mine, or with the ratio of certain constituents, such as volatile matter to total combustible. These formulas will not be given in this brief treatment.

(d) Fig. 315 gives *Mahler's Curve*,* which shows in a general

* Redrawn from curve given in U. S. Geol. Survey Professional Paper No. 48.

way how the heat value per pound of *combustible* varies with the percentage of fixed carbon present, and which also shows the range of percentages of the fixed carbon in the different kinds of coal as they are usually classified. This curve shows clearly that of all coals the semibituminous has the combustible of the highest heat value; and in connection with the map in Fig. 314, it is seen that in general the coals are of decreasing heat value the farther they are located from the main semibituminous bed. It is also true, generally, that the difficulty encountered in burning a coal efficiently increases with the distance of the mine from this same bed.

(e) Coal when mined always contains *moisture* and often takes up more afterward. Moisture is generally undesirable because it is not combustible and because it is vaporized and superheated during combustion, thus absorbing heat that might otherwise be utilized. Eastern coals, as mined, contain from one to five per cent of moisture, western coals from three to fifteen per cent, and lignites from ten to thirty per cent.

(f) *Ash* not only decreases the heat value of fuel, but it also increases the cost of transportation and handling of the coal per unit of heat produced, and in addition there is the cost of its disposal after combustion. The presence of ash also interferes with combustion, especially if it is of such composition as to form clinker. The percentage of ash in commercial coals ranges from three to fifteen ordinarily, and is usually greater in the smaller sizes than in the larger.

(g) *Sulphur*, although combustible, usually makes the fuel unsuitable for use under boilers and for many other purposes, if present in large quantities. The products of its combustion may, under certain circumstances, form acids by combining with water and these may attack the metal of boilers, etc. The presence of considerable quantities of sulphur is supposed to indicate a readily fusible ash which causes trouble in the boiler furnace, or in the "gas producer," by the formation of clinker.

(h) *Peat*, in its natural state, is a poor fuel containing a large percentage of moisture. Its value is improved by drying, but it is not yet generally used when other cheap fuels can be obtained.*

* As an indication of what may be expected when other fuels become scarcer, see Bulletin 16, U. S. Bureau of Mines, "The Use of Peat for Fuel and Other Purposes."

(i) *Lignite* is an unsatisfactory fuel when burned in furnaces, but recent investigations seem to indicate that it may be of great value for the making of "producer gas" for which there is a rapidly growing demand for power purposes.

(j) Western *bituminous* coals are a little harder to burn efficiently than eastern on account of the larger proportion of volatile matter contained. The higher the percentage of volatile matter in the coal the more difficult it is to burn it smokelessly and efficiently (see Chapter XXIX).

(k) Bituminous coals are sometimes classified as *caking* and *noncaking*. The fragments of the former kind coalesce into cakes while burning, which is desirable in the making of coke but may interfere with the supplying of air for combustion when the coal is burned upon grates.

(l) *Cannel coals* are bituminous coals which are very rich in hydrocarbons and burn like a candle, hence the name. These are used as "enrichers" in gas making.

Coals high in hydrocarbons burn with a *long flame* and are more difficult to burn efficiently than *short-flame coals*.

(m) *Semibituminous coals*, as has been shown, have the highest heat value per pound, and, as they burn with a comparatively short flame, they are the most desirable coals for use in boiler and similar furnaces.

TABLE XVI.—SIZES OF SOFT COAL—EASTERN STATES.*

Name.	Through Bars Spaced Apart.	Over Bars Spaced Apart.
	Inches.	Inches.
Lump	.....	$1\frac{1}{4}$
Nut	$1\frac{1}{4}$	$\frac{3}{4}$
Slack	$\frac{3}{4}$	.....

(n) *Anthracite coal* has an advantage over the other classes in burning smokelessly, and consequently is in great demand where smoke is not permitted. The available supply in the eastern part of the United States is rapidly diminishing and the price is, in general, higher than for other coals. To obtain the best results with anthracite it must be of uniform size. As the price decreases with the size, only the smaller grades, usually those less than $\frac{3}{4}''$ in diameter, are used for power purposes. Table XVII gives the usual classification, but unfortunately the names and sizes in some instances vary with the locality.

* For sizes in West see A.S.M.E. Power Test Code, p. 162.

TABLE XVII.—SIZES OF ANTHRACITE COAL.*

Name.	Through (Dia.).	Over (Dia.).
Chestnut.....	$1\frac{5}{8}$″	$\frac{7}{8}$″
Pea.....	$\frac{7}{8}$	$\frac{9}{16}$
Buckwheat No. 1.	$\frac{9}{16}$	$\frac{5}{16}$
Buckwheat No. 2.	$\frac{5}{16}$	$\frac{3}{16}$
Buckwheat No. 3.	$\frac{3}{16}$	$\frac{3}{32}$
Culm.....	$\frac{3}{32}$	.....
Run of mine.....	Unscreened	.....

(o) *Coal dust,* produced in mining the material, is almost always a waste product. Such dust is now successfully used in firing rotary kilns and similar apparatus, but is seldom used for ordinary power purposes although special devices for burning it under boilers and in producers have been used to a limited extent.

Coal dust and similar waste, known collectively as "culm" may represent from 10 per cent to as high as 50 per cent of all the coal recovered from the mine. It is, therefore, apparent that sooner or later some method of utilizing this waste will have to be adopted because of the decrease in the available supply of coal. Such material has been very successfully recovered in Europe by forming it under high pressure into *briquets* in which such materials as pitch, resins, wax tailings, starch, and several inorganic salts are used for "binders." These briquets can often be used more efficiently than ordinary lump fuel, because of their uniformity of size, advantageous shape, and general good behavior in the furnace; hence it often proves to be economical to purchase briquets at a slightly higher price than that asked for similar lump coal.†

228. Coke is the solid material left after driving off the volatile part of coal by heating with total or partial exclusion of air. Only certain coals yield coke of commercial value. Nearly all of the coke made is used in metallurgical processes, and but little as yet for power purposes.

* There are other sizes such as "Broken," "Egg," and "Stove" which need not be considered here as they are not commonly used in heat-power engineering. For their sizes see A.S.M.E. Power Test Code, page 15.

† For investigations relating to the manufacture and utilization of briquets under American conditions see bulletins of the U. S. Geol. Survey and of the U. S. Bureau of Mines.

Coke contains from 80 to 93 per cent of fixed carbon, from 5 to 18 per cent of ash, from 0.5 to 1.5 per cent of sulphur, and traces of volatile. It is interesting to note that the volatile is not entirely eliminated, and that, as all the ash in the coal remains in the coke, the percentage of inert matter present in the product must be greater than that in the original material.

The calorific value per pound of combustible is about the same as for carbon; that is, it is in the neighborhood of 14,600 B.t.u.

229. Wood. (a) As wood is about half moisture when felled, it must be dried before it is of much value as fuel. Air-dried wood generally has from 15 to 20 per cent of moisture, about 50 per cent of carbon, from $\frac{1}{4}$ to 2 per cent of ash, and the rest is volatile matter, largely inert. The heat value per pound of dry material is from 6600 to 9800 B.t.u.

When other cheap fuel is available wood is not generally used for power production. However, refuse from sawmills and other wood-working factories may be profitably utilized.

(b) **Charcoal** is made from wood in much the same manner that coke is made from coal. It is ordinarily used in power plants only when it is the by-product of some local process, such as the manufacture of turpentine or wood alcohol. It may contain from 80 to 97 per cent of carbon, depending on the temperature and treatment used in carbonizing it.

230. Municipal and Industrial Waste. In cities and industrial centers there is a constant accumulation of combustible waste and in some cases this material is burned directly as fuel, or it is gasified in suitable apparatus and the resultant combustible gas used as fuel. Installations of this character are still rare but are of growing commercial importance.

231. Natural Oil and Its Products. (a) *Petroleum*, or *Crude Oil*, has come into extensive use as fuel in the last twenty-five years. It is a more or less viscous, dark brown or greenish colored liquid occurring in natural reservoirs in the earth's crust. These reservoirs may be subterranean pockets, but are, in general, oil-saturated strata buried beneath other strata which are practically impervious to petroleum.

(b) Petroleum in its crude form generally has a specific gravity between 0.82 and 0.92. It is a mixture of various hydrocarbons

which are liquid at ordinary temperatures and pressures, and which hold in solution numbers of other hydrocarbons which otherwise would be gaseous or solid under existing conditions. In general, oils from one field are composed of the same hydrocarbons in about the same proportions, but each field has its own characteristic composition. American crude oils have from 82 to 87 per cent of carbon, from 12 to 15 per cent of hydrogen, and from 0 to 4 per cent of oxygen in their composition. The lower heat value per pound of crude petroleum varies from 18,000 to 22,000 B.t.u.

(c) Many of the more highly inflammable volatile components tend to distill off when the oil is brought to the earth's surface and is exposed to atmospheric conditions. These volatiles, some of which boil at temperatures as low as 80° F., are usually distilled off progressively in a refinery. The distillation products most commonly used as fuels are naphtha, gasoline, kerosene, and distillate, given in order of decreasing inflammability and increasing density and distillation temperature. The greater part of the remainder can be sold as " fuel oil."

(d) *Gasoline* is the name given to the group of hydrocarbons which distill off at temperatures between 150 and 300° F. Gasolines of various specific gravities are obtained by fractionating the material obtained between these temperatures, the lower gravities corresponding to the lower temperatures. The commonest grades range from 74 to 64 degrees gasoline as measured by a Baumé hydrometer. The corresponding specific gravities are 0.686 and 0.722. The relative proportions of carbon and hydrogen in gasolines are roughly 85 per cent and 15 per cent, and the lower calorific value is about 19,200 B.t.u. per pound. The flash point of gasoline, that is, the temperature at which readily inflammable vapors are given off from an exposed surface, is generally well below 70° F.

(e) *Kerosene* is the name of the next important group of hydrocarbons which distill over after the gasolines. Their specific gravity is from 0.78 to 0.82 and the flash point is from 70° to about 150° F. for the different grades. The B.t.u. per pound of kerosene is about 18,500 lower value.

(f) *Fuel Oil,* having little highly volatile matter, can be handled without danger and, being very cheap, is quite widely used under boilers and in furnaces. The lower calorific value per pound is

extremely variable, but may be taken roughly at 18,000 B.t.u. per pound.

(g) The higher calorific value of U. S. petroleum and its distillates, ranging from crude oil to gasoline, varies quite regularly with the specific gravity of the material, and is expressed approximately by the following formula,* which may be assumed correct within 2 per cent,

$$\text{B.t.u. per pound} = 18{,}650 + 40\,(B - 10), \quad . \quad . \quad (335)$$

in which B = degrees on the Baumé hydrometer. Since the Baumé scale increases as the density of the material becomes less, this formula indicates that the lighter distillates have greater heat values than the heavier ones, when figured on a weight basis. The reverse is true for heat value per *gallon*, a unit commonly used with liquid fuels; hence a barrel of light petroleum distillates of any kind will, in general, have less heat value than a barrel of heavier distillates or of the original oil free from water.

232. Alcohol. (a) Both *Methyl* ("Wood") and *Ethyl* ("Grain") alcohol are used as fuel to a limited extent. Methyl alcohol (CH_4O) is poisonous and is produced during the dry distillation of wood. Ethyl alcohol (C_2H_6O) is made, by a fermentation and distillation process, from grain, fruit, or vegetable matter containing starch or sugar.

(b) The material known as *Denatured Alcohol* consists of ethyl alcohol with the addition of from 1 to 10 per cent of Methyl alcohol and other substances which prevent its use in beverages and give it an unpleasant odor. Commercial alcohol generally also contains 10 per cent or more of water by volume.

Denatured alcohol has many theoretical and practical advantages over gasoline as a fuel for certain purposes, but at present its relative cost, in this country at least, is so great as to prevent its extensive use.

(c) The higher calorific value of *Absolute* Ethyl alcohol (i.e. containing no water) is about 13,000 B.t.u./lb.; the lower value is about 12,000 B.t.u./lb. The heat value of "Commercial" alcohol, containing about 10 per cent of water, by volume, varies with the materials used in denaturizing. The lower heat value is generally near 10,500 B.t.u./lb.

* H. C. Sherman and A. H. Kropff, Jour. Am. Chem. Soc., Oct., 1908.

233. Natural Gas. (a) This material is found in various places, but particularly in certain regions of the United States, either escaping through cracks and faults in the earth's crust, or held at high pressure in huge underground reservoirs which may be tapped by drilling wells similar to those used for obtaining oil; in fact, most oil wells yield a certain amount of natural gas.

(b) Natural gas is a mixture of combustible and incombustible gases, the latter generally occurring in very small quantities. The proportions and even the constituents of the gas are seldom the same in different districts and occasionally vary unaccountably even in the same well.

The principal combustible constituents are Methane (CH_4), and Hydrogen (H_2). The former generally occurs in far greater proportion than the latter. The other combustible gases which usually occur in very small proportions are Carbon Monoxide, (CO), sulphur compounds such as Hydrogen Sulphide (H_2S), and certain hydrocarbon gases, such as Ethylene (C_2H_4), and others.

The principal incombustible constituents are generally small proportions of Carbon Dioxide (CO_2), Nitrogen (N_2), and Oxygen (O_2), if this latter may be considered an incombustible.

(c) Natural gas is an ideal form of fuel for many industrial purposes, and is readily piped distances of one hundred miles and more for use in industrial centers far from a natural supply. Unfortunately many of the wells are becoming exhausted and the price is rising in proportion.

TABLE XVIII.* — TYPICAL ANALYSES OF NATURAL GAS.

Location of Field.	Analyses in Volumes, Per Cent.								
	Hydrogen, H_2.	Methane, CH_4.	Ethylene, C_2H_4.	Illuminants, C_6H_6, etc.	Carbon dioxide, CO_2.	Carbon monoxide, CO.	Oxygen, O_2.	Nitrogen, N_2.	Sulphuretted hydrogen, H_2S.
Anderson, Ind.	1.86	93 07	0.47	.....	0.26	0.73	0.42	3.02	0.15
Louisville, Ky.	1.31	87.75	.....	.....	6.60	.....	.....	4.34	.....
Olean, N. Y.	.....	96 50	.....	1.00	.....	0.50	2.00	.....	.....
Findlay, Ohio	2.18	92.60	.....	0.31	0.26	0.50	0.34	3.61	0.20
Harvey Well, Pa.	13 50	80.11	5.72	.....	0 66	.....	.....	.....	.....
Creighton, Pa.	.....	96.34	.....	trace	3.64	.....	.....	.....	.....
Pittsburgh, Pa.	20.02	72.18	.....	6.30	0.80	1.00	0.80	.....	.....
Pechelbrown, Germany	.....	77.03	.....	4.80	3.60	3.50	1.80	8.90	.....
Aspharon Peninsula, Russia	0.34	92.49	.....	4.11	0.93	.....	.....	2.13	.....

* Abstracted from Table in "Calorific Value of Fuels," Herman Poole, p. 241.

Table XVIII gives some typical analyses of natural gas from several different districts. The lower calorific value generally varies from about 950 to 1000 B.t.u. per cubic foot.

234. Artificial Gases. (a) The principal artificial gases are made from coal or crude oil, but there are also many processes for producing combustible gases from vegetable and animal by-products and wastes. Many of the latter are successful in isolated cases but they are not yet of great commercial importance.

(b) Most of the artificial gases are made either by destructive distillation, by partial combustion, by chemical decomposition, or by various combinations of these processes.

Destructive distillation occurs when the gas-making material is heated in a chamber from which air is more or less perfectly excluded. Illustrations of gases made by this process are "Illuminating," "Retort," or "Town Gas," used for illumination, and gas made in "By-product" or "Retort Coke Ovens" used for illumination and power.

"Producer Gases" are the best examples of those which in theory are made by a process of incomplete combustion. Practically this is always more or less combined with chemical decomposition. These gases have become so important of late in connection with the internal combustion engine that they will be discussed later in a separate chapter.

(c) The use of artificial gases as fuel in internal combustion engines results generally in a greater output of available energy than would the use of the solid fuels, from which the gases are made, in other heat-power apparatus; hence these gases may be expected to become more and more important with the depletion of the natural stores of fuel and with the growth of the spirit of conservation of the earth's resources.

CHAPTER XXVIII.

COMBUSTION.

235. Definitions. (a) To the engineer *Combustion* means the chemical combination of certain elements with oxygen at such a rate as to cause an appreciable rise of temperature.

Practically all chemical reactions are accompanied by liberation or absorption of heat. When heat is liberated the reaction is called *exothermic;* when heat is absorbed the reaction is called *endothermic.*

(b) During these reactions, with other conditions constant, (1) the amount of heat energy liberated or absorbed is independent of the time occupied; and (2) for any material taking part in the reaction, the heat change is directly proportional to the mass of that material.

(c) Materials which can be caused to unite with oxygen to produce heat are known as *Combustibles.* For engineering purposes they are limited to Carbon and Hydrogen; these, either pure or in various combinations, constitute practically the entire stock of available combustibles, although a trace of sulphur usually appears as an impurity.

(d) In heat-power engineering the object of combustion is either the production of heat directly, or the formation of a more suitable kind of combustible, such as gas or coke, from the original material.

Useful combustion data are given in Table XIX. In it the values of specific densities and volumes are given for an average atmospheric temperature of 62° F. as well as for 32° F.

236. Combustion of Carbon. (a) Carbon is the principal combustible in nearly all engineering fuels. This element combines with oxygen to form two oxides, — Carbon monoxide (CO), and Carbon dioxide (CO_2). If CO is formed, the combustion is said to be "incomplete"; if CO_2 is formed, it is said to be "complete" or "perfect."

TABLE XIX. — COMBUSTION DATA.

Name.	Symbol.	Atom. or Mol. Wt.	Spec. Wt. (Gas). 32° F., 14.7 lbs. per sq. in.	Spec. Wt. (Gas). 62° F., 14.7 lbs. per sq. in.	Spec. Vol. (Gas). 32° F., 14.7 lbs. per sq. in.	Spec. Vol. (Gas). 62° F., 14.7 lbs. per sq. in.	Spec. Heat (Gas). C_p	Spec. Heat (Gas). C_v	Combustion Data per Lb. of Combustible. Symbol of Prod.	Theor. O in lbs.	Theor. Air. Lbs.	Theor. Air. Cu. ft. at 62°–14.7	Calorific Value, B.t.u.	Theor. Temp. Rise, using Theor. Air. Pres. Const.	Theor. Temp. Rise, using Theor. Air. Vol. Const.
Oxygen	O	16	0.0892	0.0840	11.21	11.90	0.2175	0.1551							
Nitrogen	N	14	0.0781	0.0735	12.81	13.60	0.2438	0.1735							
Carbon	C	12							CO	1.33	5.78	74.6	4,500	2940	4030
									CO_2	2.67	11.40	150	14,600	5080	7080
Hydrogen	H	1	0.005618	0.00529	178.0	188.9	3.410	2.436	H_2O	8	34.16	449	62,000 / 52,000	5010	6800
Sulphur	S	32							SO_2	1	4.27	56.1	4,000	3620	4920
Carbon monoxide	CO	28	0.0781	0.0735	12.81	13.6	0.2425	0.1734	CO_2	0.571	2.44	32.1	4,300	5580	7570
Carbon dioxide	CO_2	44	0.1224	0.116	8.102	8.62	0.2008	0.1548							
Water	H_2O	18													
Sulphur dioxide	SO_2	64	0.1828	0.1722	5.471	5.81	0.1544	0.1229							

(b) The reactions occurring during combustion may be expressed by chemical equations, the symbols used standing for definite proportions by mass or weight. For engineering purposes the atomic weight of Carbon may be taken as 12, that of Nitrogen as 14, and that of Oxygen as 16.

(c) When carbon and oxygen combine to form *Carbon Dioxide* the reaction is expressed by

$$C + O_2 = CO_2; \quad . \quad . \quad . \quad . \quad . \quad . \quad (336)$$

the *weights* combined are

$$12 \text{ of } C + (2 \times 16) \text{ of } O = 44 \text{ of } CO_2,$$

and dividing this by 12 gives

$$1 \text{ of } C + 2\tfrac{2}{3} \text{ of } O = 3\tfrac{2}{3} \text{ of } CO_2. \quad . \quad . \quad . \quad (337)$$

Thus if 1 pound of carbon unites with $2\frac{2}{3}$ pounds of O the result is $3\frac{2}{3}$ pounds of CO_2. It is also found that *heat* equal to about 14,600 B.t.u. is *liberated* per pound of carbon when this reaction occurs.

(d) When carbon is burned to *Carbon Monoxide*, the reaction is expressed by

$$2\ C + O_2 = 2\ CO; \quad . \quad . \quad . \quad . \quad . \quad . \quad (338)$$

the *weights* combined are

$$(2 \times 12) \text{ of } C + (2 \times 16) \text{ of } O = 56 \text{ of } CO,$$

and dividing this by 24 gives

$$1 \text{ of } C + 1\tfrac{1}{3} \text{ of } O = 2\tfrac{1}{3} \text{ of } CO. \quad . \quad . \quad . \quad (339)$$

The *heat liberated* is, in this case, about 4500 B.t.u. per pound of carbon.

(e) The *gaseous CO* formed as above can be burned to CO_2. The reaction is

$$2\ CO + O_2 = 2\ CO_2; \quad . \quad . \quad . \quad . \quad . \quad (340)$$

the *weights* combined are

$$\{2 \times (12 + 16)\} \text{ of } CO + (2 \times 16) \text{ of } O = 88 \text{ of } CO_2,$$

and dividing this by 24 gives

$$2\tfrac{1}{3} \text{ of } CO + 1\tfrac{1}{3} \text{ of } O = 3\tfrac{2}{3} \text{ of } CO_2. \quad . \quad . \quad . \quad (341)$$

Thus the $2\frac{1}{3}$ pounds of CO, which would result from the combination of 1 pound of carbon as in Eq. (339), would combine with $1\frac{1}{3}$ pounds of O to form $3\frac{2}{3}$ pounds of CO_2.

This reaction is accompanied by the *liberation of heat* equal to about 10,100 B.t.u. per pound of *carbon*. Therefore the heat

liberated *per pound of carbon monoxide gas* must be $10100/2\frac{1}{3} = 4300$ B.t.u.

The specific volume of *CO* is 12.81 cu. ft. at 32° F. and 14.7 pounds pressure. Hence the heat liberated *per cu. ft. of CO* under these conditions is $4300 \div 12.81 = 335$ B.t.u. As *CO* is a constituent of many commercial fuel gases and as these are usually measured volumetrically instead of gravimetrically, this value is convenient for determining the heat available due to the *CO* present.

(f) Equations expressing both the reaction and the liberation of heat *per pound of material** may be written as follows:

$$C + O_2 = CO_2 + (14{,}600 \text{ B.t.u. per pound } C) \quad . \quad . \quad (342)$$

$$2\,C + O_2 = 2\,CO + (4500 \text{ B.t.u. per pound } C) \quad . \quad . \quad (343)$$

$$2\,CO + O_2 = 2\,CO_2 + \begin{cases} (10{,}100 \text{ B.t.u. per pound } C) & (344) \\ \text{or } (4300 \text{ B.t.u. per pound } CO) & (345) \end{cases}$$

Noting that $4500 + 10{,}100 = 14{,}600$, it is evident from these equations that when carbon is burned to CO_2, the ultimate results are the same whether the process takes place in one or in two steps.

Further, if part of the pound of carbon (say C_x pounds) is burned to CO_2 and the rest (C_y pounds) to *CO*, the heat liberated is

$$\text{B.t.u.} = 14{,}600\,C_x + 4500\,C_y. \quad . \quad . \quad . \quad (346)$$

(g) If heat is the object of combustion, the carbon should of course be burned to carbon dioxide rather than to carbon monoxide. If *CO* is formed instead of CO_2, the proportion of the heat lost is $(10{,}100/14{,}600) = .69 +$, or about 70 per cent. If, however, the *CO* is burned later, the rest of the heat may be recovered.

* As will be seen later, it is sometimes more convenient to modify Equation (342) so that the heat quantity as well as symbol *C* will correspond to 12 lbs. of carbon (the pounds being taken as numerically equal to the molecular weight of *C* involved). The equation then becomes

$$C + O_2 = CO_2 + 175{,}200 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (342a)$$

where $175{,}200 = 12 \times 14{,}600$.

Similarly, for the 24 lbs. of carbon represented by 2 C and by 2 CO, Eqs. (343) and (344) become

$$2\,C + O_2 = 2\,CO + 108{,}000 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (343a)$$

and

$$2\,CO + O_2 = 2\,CO_2 + 242{,}400 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (344a)$$

in which $108{,}000 = 24 \times 4500$, and $242{,}400 = 24 \times 10{,}100$.

When there is less than enough oxygen to burn the carbon to carbon dioxide, both CO_2 and CO will be formed and the relative amounts of each can be determined in the following manner: Assume first that there is 1 per cent deficiency in the oxygen supply needed to form CO_2 and that in consequence 99 per cent of the carbon is burned to CO_2 and 1 per cent remains C; then assume that this 1 per cent of C combines with some of the CO_2 according to the equation $C + CO_2 = 2\ CO$; thus it is seen that there is finally 2 per cent of the carbon present in CO and 98 per cent in CO_2.* In general, then, if there is y per cent deficiency of oxygen there will be $2y$ per cent of the carbon burned to carbon monoxide instead of to CO_2. Hence from the preceding paragraph it follows that with y per cent deficiency of oxygen there is $(2 \times .7)\ y$ per cent, or $1.4\ y$ per cent less heat developed than if all the carbon were burned to CO_2. The great importance of having a sufficient supply of oxygen is thus apparent.

The discussion in the preceding paragraph presupposes that the oxygen supply is at least sufficient to burn all the carbon to CO, — that is, that the deficiency is not more than 50 per cent on a basis of combustion to CO_2. Should y be greater than 50 per cent, some of the carbon will not be burned at all. The percentage not affected will be $2\ (y - 50)$.

(h) It is sometimes possible to *reverse chemical reactions*, — in the present case, for instance, to break up one of the oxides into the original elements. When this is done the same amount of heat will be absorbed during the decomposition as was originally liberated during combination.

Thus, if it is possible to break up the quantity of CO_2 containing a pound of carbon into the elements C and O, it will require an expenditure of 14,600 B.t.u. Similarly, it will require 10,100 B.t.u. to reduce to the monoxide CO an amount of CO_2 containing one pound of carbon, and 4500 B.t.u. per pound of carbon will be consumed in separating CO into its elements.

237. Weights of Oxygen and Air Necessary for Combustion of Carbon. It was shown that for each pound of carbon burned to CO_2 there are required $2\frac{2}{3}$ pounds of oxygen, or $1\frac{1}{3}$ pound of

* These statements should be limited to take account of certain " equilibrium " conditions which will be discussed in a later chapter. For the present purpose, however, they are sufficiently exact.

O if CO is formed. If, as before, C_x and C_y are the weights of carbon burned respectively to CO_2 and to CO, then the number of pounds of oxygen used are

$$\text{Pounds of } O = 2\tfrac{2}{3}\, C_x + 1\tfrac{1}{3}\, C_y. \quad . \quad . \quad . \quad (347)$$

TABLE XX. — PROPERTIES OF AIR.

	Relative Proportions.		Ratio of Air to O.		Ratio of N to O.		Spec. Wt. at Atm. Pres.		Spec. Vol. at Atm. Pres.	
	Exact.	Approx.	Exact.	Approx.	Exact.	Approx.	32°	62°	32°	62°
By Weights.	0.766 N 0.234 O	0.77 N 0.23 O	4.27	4.35	3.27	3.35	0.08072	0.07609	12.39	13.14
By Volumes.	0.791 N 0.209 O	0.79 N 0.21 O	4.78	4.76	3.78	3.76	Specific Heats.			
							C_p=0.238		C_v=0.169	

Since, in ordinary engineering work, pure oxygen cannot, in general, be conveniently obtained, it is customary to utilize the oxygen of the atmosphere. Table XX shows that air is composed by *weight* of about 23 parts of oxygen and 77 of nitrogen;* thus the ratio of air to the oxygen it contains is about $100/23 = 4.35$, and of nitrogen to oxygen is about $77/23 = 3.35$. Hence, for each pound of oxygen supplied there must be used 4.35 pounds of air containing 3.35 pounds of inert nitrogen. An equation for finding the weight of air required to burn C_x pounds of carbon to CO_2 and C_y pounds to CO can therefore be found by multiplying both sides of Eq. (347) by 4.35. This gives (approximately)†

$$\text{Pounds of Air} = 11.6\, C_x + 5.8\, C_y. \quad . \quad . \quad . \quad (348)$$

238. Volumes of Gases Involved in Combustion of Carbon. (a) The combustion formulas, like other chemical formulas involving gases, can be read in terms of molecules and of volumes as well as in terms of weights. Thus

$$2\, C + O_2 = 2\, CO$$

may be read, "two atoms of carbon unite with one molecule of oxygen to form two molecules of carbon monoxide." But accord-

* This is the common engineering assumption. Atmospheric air always contains carbon dioxide and water vapor as well as a few rare gases such as argon.

† The quantity 11.6 (often taken as 12), representing the weight of air required for complete combustion of one pound of C, will be frequently used by the student hereafter; hence it should be remembered.

ing to *Avogadro's law* the same number of molecules are contained in equal volumes of all the different gases when at the same temperature and pressure. Therefore, since every molecule of oxygen in a given volume of gas is capable of yielding two molecules of carbon monoxide, it follows that one volume of oxygen will yield two volumes of CO, both being measured at the same temperature and pressure.

Similarly the equation

$$C + O_2 = CO_2$$

shows that one volume of oxygen yields one volume of carbon dioxide; and the equation

$$2\ CO + O_2 = 2\ CO_2$$

shows that two volumes of carbon monoxide combine with one volume of oxygen to form only two volumes of carbon dioxide.

(b) In the first case cited there was an increase of gas volume, in the second there was no change, and in the third there was a diminution. If the gases appear in terms of molecules (O_2, N_2, H_2, CO, *etc.*) in the chemical equations, the *coefficients of the molecule symbols represent relative volumes.*

(c) Since air is composed of about 21 parts of oxygen and 79 parts of nitrogen by volume, every volume of atmospheric oxygen will carry with it $\frac{79}{21} = 3.76$ volumes of nitrogen.

When air is used to support combustion the nitrogen takes no chemical part in the reactions considered but simply mixes with the products of the combustion and is known as a *diluent.**

A simple relation can now be shown: Since the volume of CO_2 formed by combustion of carbon equals the volume of the oxygen used in the process and since oxygen forms 21 parts of air by volume, it follows that with " complete " combustion the products will consist of 21 parts CO_2 and 79 parts N by volume when not diluted by the presence of excess air.

(d) Then if the carbon is burned in air and if analysis of the " flue " gas shows that the volume of CO_2 is less than **21 per cent,** it follows that either (1) there is more air present than is required for complete combustion, the excess acting as a diluent, or (2) that there is a deficiency of air, with the result that only a part

* Under some conditions part of the nitrogen burns to an oxide, but the quantity thus consumed is small in all the ordinary engineering processes.

of the carbon is burned to CO_2, the rest appearing in CO. If the percentage of CO_2 is less than 21 and no CO is found in the flue gas it indicates that there is excess air; and if CO is present there is a deficiency.*

(e) The *percentage of CO_2 by volume* in the flue gas mixture can be computed for any condition of combustion, by using the following formula,

$$\text{Per cent of } CO_2 \text{ by vol.} = \frac{\text{vol. of } CO_2}{\text{total vol. of gas}} \times 100$$

$$= \frac{w\mathbf{V}}{(w\mathbf{V}) + (w\mathbf{V})_1 + \text{etc.}} \times 100 = \frac{w\mathbf{V}}{\Sigma\,(w\mathbf{V})_n} \times 100, \quad (349)$$

in which w = weight of CO_2 present in the mixture,
w_1, w_2, etc. = weights of the other gases present,
$\mathbf{V}$ = specific volume of CO_2,
$\mathbf{V}_1$, $\mathbf{V}_2$, etc. = specific volumes of the other gases.

The specific volumes of gases are given in Table XIX.

(f) If an *excess of air* is supplied, say x per cent more than is required for perfect combustion, the maximum per cent of CO_2 by volume which could be present in the flue gas can be computed in the following manner, based on the combustion of *one pound of carbon*. In this case the combustion of the pound of carbon will result in $3\frac{2}{3}$ pounds of CO_2; it will theoretically require 11.6 pounds of air; there will be $0.77 \times 11.6 = 8.9$ pounds of nitrogen accompanying the oxygen used for combustion; and there will be $\left(11.6 \times \frac{x}{100}\right)$ pounds of excess air. The following table gives the weights (w) of gas present per pound of C burned to CO_2, their specific volumes ($\mathbf{V}$) at 62° F., and the products of these quantities ($w\mathbf{V}$).

TABLE XXI. — FLUE GAS CONSTANTS.

Gas.	w	$\mathbf{V}_{62}$	$w\mathbf{V}$
CO_2	$3\frac{2}{3}$	8.62	31.6
$N_{(\text{theoretical})}$	8.9	13.60	121.0
Air	$\frac{11.6\,x}{100}$	13.14	$1.52\,x$

* These statements should be limited to take account of certain "equilibrium" conditions discussed in a later chapter. For the present purpose, however, they are sufficiently exact.

The sum of the last column is,

$$\Sigma\,(wV)_n = 31.6 + 121.0 + 1.52\,x = 152.6 + 1.52\,x$$
$$= 153\left(1 + \frac{x}{100}\right), \text{ approx.}$$

Then from Eq. (349)

$$\text{Per cent } CO_2 \text{ by vol.} = \frac{31.6}{153\left(1 + \frac{x}{100}\right)} \times 100, \text{ approx.,} \quad (350)$$

which gives the proportion of CO_2 in the flue gas.

(g) Again, since the per cent of CO_2 by volume decreases directly as the quantity of total air is increased, it is evident from (c) that

$$\text{Per cent } CO_2 \text{ by volume} = 21 \div (1 + x/100), \quad (351)$$

in which x, as before, represents percentage of excess air. Simplifying Eq. (350) results in the same equation (approximately). The relation of the CO_2 to x is shown in Fig. 316 by the curve to the right of oO.

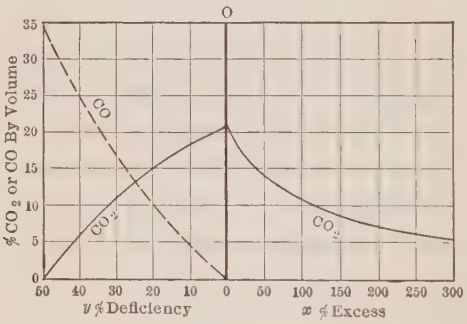


Fig. 316.

(h) The expression $(1 + x/100)$ is known as the **excess coefficient** X. In words, — the excess coefficient is the number by which the theoretical amount of oxygen, or air, required must be multiplied to find that actually supplied.

If the CO_2 percentage is known, and if the combustion is complete (and only in that case) the **per cent of excess air** x, can be found from Eq. (351). Thus

$$x = \left(\frac{21}{\text{per cent } CO_2} - 1\right)100. \quad (352)$$

(i) For the case of complete combustion, the percentage of

excess air can also be found when both the total volume (N) of nitrogen and the volume (O_x) of that part of the oxygen which remains free after combustion are known. The volume of nitrogen accompanying the excess oxygen is $79/21 \times O_x = 3.76\ O_x$, and that corresponding to the oxygen used in combustion is $N - 3.76\ O_x$. Hence, since the nitrogen undergoes no change, the percentage of excess air is

$$x = 100\,(3.76\ O_x) \div (N - 3.76\ O_x), \quad . \quad . \quad (353)$$

and the excess coefficient is

$$X = 1 + \frac{x}{100} = 1 + \frac{3.76\ O_x}{N - 3.76\ O_x}. \quad . \quad . \quad . \quad (354)$$

(j) In most cases it is possible to have *incomplete combustion* of part of the carbon although *sufficient air* is present, since the air may not be properly distributed. In such cases x and X can be determined if the volume per cent of CO is known in addition to the N and O. Since each volume of CO present could have combined with half its volume of oxygen to form CO_2, it follows that, on the basis of complete combustion, the excess oxygen is equal to $O - CO/2$. The nitrogen accompanying this is $3.76\,(O - CO/2)$, and that corresponding to the oxygen required for complete combustion is $N - 3.76\,(O - \frac{1}{2}\,CO)$. The percentage of excess air for this case is, then

$$x = \frac{3.76\,(O - \frac{1}{2}\,CO) \times 100}{N - 3.76\,(O - \frac{1}{2}\,CO)}, \quad . \quad . \quad . \quad . \quad . \quad (355)$$

and the excess coefficient is

$$X = 1 + \frac{x}{100} = 1 + \frac{3.76\,(O - \frac{1}{2}\,CO)}{N - 3.76\,(O - \frac{1}{2}\,CO)}. \quad . \quad . \quad (356)$$

(k) In case there is a *deficiency of air* amounting to y per cent, there is $2y$ per cent of the carbon burned to CO, as has already been shown (in Sect. **236 g**); in burning one pound of carbon there will result $2y/100 \times 2\frac{1}{3}$ pounds of CO, $(1 - 2y/100) \times 3\frac{2}{3}$ pounds of CO_2, and the nitrogen present will be $(1 - y/100) \times (0.77 \times 11.6)$ pounds. Tabulating these values gives:

TABLE XXII. — FLUE GAS CONSTANTS.

Gas.	w	V_{62}	wV
CO_2	$(1-2y/100)\times 3\frac{2}{3}$	8.62	$31.6\ (1-2y/100)$
CO	$(2y/100)\times 2\frac{1}{3}$	13.60	$31.7\ (2y/100)$
N	$(1-y/100)\times 8.9$	13.60	$121.0\ (1-y/100)$

The summation of the last column gives

$$\Sigma\,(wV)_n = 152.6 - 1.21\,y.$$

Then, from Eq. (349),

$$\text{Per cent } CO_2 \text{ by vol.} = \frac{31.6\,(1 - 2\,y/100)}{152.6 - 1.21\,y} \times 100 \quad . \quad (357)$$

and

$$\text{Per cent } CO \text{ by vol.} = \frac{31.7\,(2\,y/100)}{152.6 - 1.21\,y} \times 100. \quad . \quad . \quad (358)$$

The relation of the percentage volumes of CO_2 and CO in the flue gas to the percentage deficiency of air, is shown in Fig. 316 by the curves to the left of oO.

239. Temperature of Combustion. (a) When combustion occurs, the heat energy liberated tends to be dissipated. But if the combustion can be imagined to occur within a vessel perfectly impervious to heat, then all the liberated heat must remain within the vessel, and the products of combustion would be raised to a high temperature. This temperature is known as the *theoretical temperature of combustion* and is readily calculated.

(b) First, assuming only one gas as the product of combustion, and no heat absorbed by the surrounding vessel, the *theoretical temperature rise* will be obtained by dividing the available heat by the product of the weight and the specific heat of the gas formed. If the vessel is assumed not to change in size, the specific heat for constant volume must be used; but if the vessel is the equivalent of one fitted with a movable piston so arranged as to maintain constant pressure, the specific heat would be that for constant pressure.

For example take the reaction

$$2\,C + O_2 = 2\,CO + (4500 \text{ B.t.u. per pound of } C).$$

Since $2\frac{1}{3}$ pounds of CO are formed per pound of carbon, the theoretical *rise* of temperature with the theoretical supply of pure oxygen will be

$$Rise = \frac{4500}{C_n \times 2\frac{1}{3}},$$

in which C_n is either the specific heat at constant volume (C_v), or at constant pressure (C_p), as the case may be.

(c) If air is used to furnish the oxygen, the nitrogen must also be heated. Then, since the weight of N, accompanying the O

used per pound of C, is $1\frac{1}{3} \times 77/23 = 4.47$, the resulting temperature *rise* is

$$Rise = \frac{4500}{C_n \times 2\frac{1}{3} + C_n' \times 4.47},$$

in which the primed specific heat is for the nitrogen, and the other for CO as before. Similar equations may be written for other combustibles and other products of combustion.

(d) The *temperature theoretically attained* will be the sum of temperature t_0 existing before the start of the reaction and the temperature rise as found above.

In general, for any number of products of combustion,

$$t = t_0 + \frac{\Delta Q}{C_0 w_0 + C_1 w_1 + C_2 w_2 + C_3 w_3 + \text{etc.}}$$

$$= t_0 + \frac{\Delta Q}{\Sigma (Cw)_n}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (359)$$

in which

t_0 = initial temperature,

ΔQ = heat liberated,

C_0, C_1, C_2 = the specific heats of the products of combustion, at constant volume or pressure, as the case may be, and

w_0, w_1, w_2 = the weights of the products of combustion.

The specific heats of the products of combustion * are given in Table I on pages 40 and 41.

(e) It is evident that the larger the denominator of the fraction in Eq. (359) the lower will be the theoretical temperature of combustion. If, then, an inert gas such as nitrogen is carried through the heating process, as when air is used instead of pure oxygen, the theoretical temperature will be lowered; and should more air be supplied than is needed for complete combustion, the temperature will be still further reduced. Diluents, though taking no essential chemical part in the reaction, thus play a very important part from the physical side; they always reduce the theoretical maximum temperature attainable by the combustion.

The way the *temperature rise* theoretically varies with excess or deficiency of air, when carbon is burned at constant pressure, is shown in Fig. 317.

(f) If any heat is lost during the period of combustion, as,

* See also (g) of this section.

for instance, by radiation, the numerator of Eq. (359) will be diminished by the amount lost and the theoretical temperature rise will, of course, be decreased. This indicates the advisability of causing combustion to take place as rapidly as possible, because, despite the fact that the same amount of heat is liberated during slow as during rapid combustion, the actual time allowed for radiation in the latter case is less, and hence, other things being equal, the temperature attained will be greater.

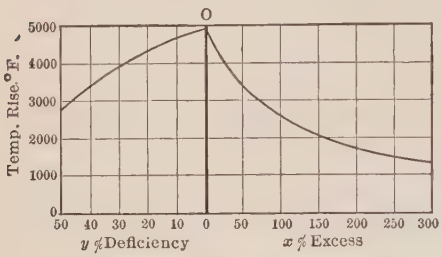


Fig. 317.

(g) In the chapters on the theory of the ideal gas it was stated that for ordinary work the specific heats of real gases might be considered constant. They are, however, really not constants, and some of them increase rapidly in value at high temperatures. From the previous consideration of the behavior of superheated steam this is just what would be expected.

There is still considerable lack of agreement between the values of the high temperature specific heats of gases as determined by different investigators. The later work in this field does, however, give sufficiently concordant results to definitely settle the fact that such increase with the temperature does occur, and to warrant the engineer in assuming some of the values obtained as sufficiently accurate for practically all engineering calculations.

The values for mean specific heats given in Fig. 318, taken from curves prepared by Prof. G. B. Upton and published in "Experimental Engineering" by Carpenter and Diederichs, represent the most probable values for the different gases named. It will be noted that no account is apparently taken of variation of the specific heat with pressure. Such variation is negligible for most gases with which the engineer ordinarily deals, — the principal exceptions being water vapor and ammonia. For water vapor the variation with pressure is rapid at low temperatures but becomes less so at higher temperatures, as was shown in Fig. 40.

(h) Since the specific heat increases with temperature, the value of the denominator of Eq. (359) must increase when the numerator becomes greater, other things being equal; thus the higher the temperature the less effective is a given quantity of heat in causing a rise of one degree. It follows that in practical work, even if all radiation could be prevented, the temperature will never rise as high as Eq. (359) would indicate when the ordinary values of the specific heat are used.

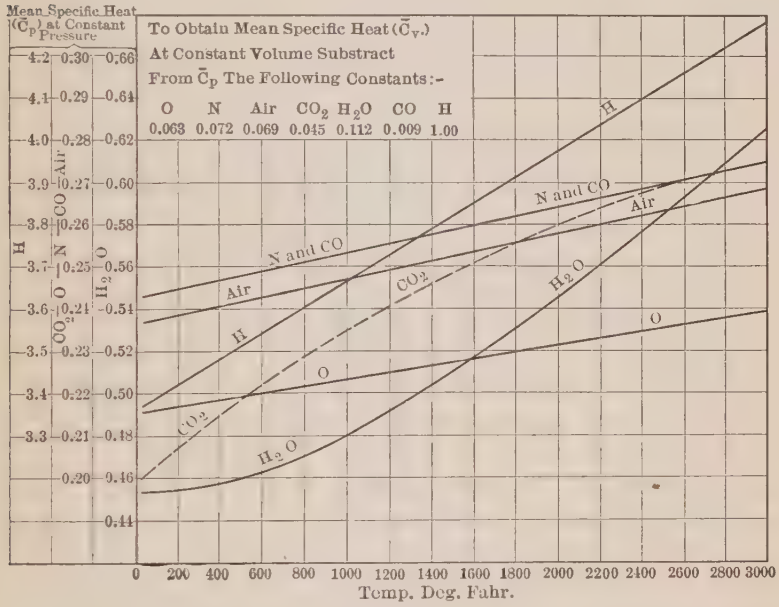


Fig. 318.

To determine the *real* temperature rise (neglecting radiation loss) for any given heat supply, the *mean* specific heats must be used in the denominator of the equation. The only satisfactory way of doing this would be to guess at the temperature expected, choose the corresponding mean specific heats, determine the resultant temperature and compare with the value assumed. If the difference is great a closer approximation can be made, and so on. This is similar to many of the calculations used in connection with superheated steam.

240. Combustion of Hydrogen. (a) Hydrogen burns according to the following equation:

$$2\ H_2 + O_2 = 2\ H_2O. \quad . \quad . \quad . \quad . \quad . \quad . \quad (360)$$

The weights combined are

$$(2 \times 2)\ of\ H + (2 \times 16)\ of\ O = 36\ of\ H_2O,$$

and dividing this by 4 gives

$$1\ of\ H + 8\ of\ O = 9\ of\ H_2O. \quad . \quad . \quad . \quad . \quad (361)$$

Then, for each pound of hydrogen burned, 8 pounds of oxygen must be supplied, and 9 pounds of water will result. It is also found that about 62,000 B.t.u. are liberated per pound of hydrogen burned.

(b) Problems involving the combustion of hydrogen are often complicated by the fact that many of the real combustibles contain some oxygen, which may exist as a constituent of a $C_xH_yO_z$ compound, or in combination with some of the hydrogen as H_2O, or in a number of other different ways. Obviously, to calculate the heat that would be liberated by a pound of such material would require a knowledge of the condition of all oxygen present; but unfortunately such knowledge is seldom available, hence it is customary to consider that all oxygen present is combined with hydrogen as H_2O and that only the remainder of the hydrogen can burn to liberate heat. This combustible part of the total hydrogen is known as *available* or *uncombined hydrogen.*

According to the assumption just given, the "available" hydrogen can be determined in any case by subtracting from the total hydrogen the amount which could be combined with all the oxygen present. Equation (361) above shows that a given weight of oxygen could be combined with one-eighth its weight of hydrogen and it follows from this that the weight of the unavailable hydrogen must be one-eighth of the weight of oxygen present in a compound.

Then if H represents the total weight of hydrogen present and if there are O pounds of oxygen which are assumed to be already combined with part of this hydrogen, the available hydrogen must weigh $(H - O/8)$ pounds. The oxygen required for the combustion of the available hydrogen is

$$\text{Pounds of } O = 8\,(H - O/8), \quad . \quad . \quad . \quad . \quad (362)$$

and the weight of air required to supply this oxygen is found by multiplying this equation by 4.35; thus

$$\text{Pounds of air} = 34.8\,(H - O/8) \text{ (approx.)}. \quad . \quad (363)$$

(c) If hydrogen at 60° F. is burned to H_2O and the latter is afterwards cooled to 60°, the quantity of heat obtained varies according to the conditions of cooling. If the material is contained in a vessel equivalent to a cylinder closed with a movable piston exerting a constant pressure and if at the end of the cooling process all the water exists as liquid, a certain amount of heat equal to about 62,000 B.t.u. (experimental value 61,950) per pound of available hydrogen will be obtained. In this book this will be called the "**higher heat value**" of hydrogen.

On this basis the higher heat value may be defined as the quantity of heat obtained when the products of combustion are cooled in such manner that the water vapor resulting from the combustion of one pound of hydrogen (initially at 60° F.) is completely condensed at constant pressure to a liquid at a temperature of 60° F. Then, when a combustible containing H pounds of hydrogen and O pounds of oxygen is burned, the heat obtained from the available hydrogen, on the basis of this higher heat value, would be

$$B.t.u. = 62{,}000\,(H - O/8).$$

(d) Other definitions of higher heat value are sometimes given. Instead of cooling to 60°, some other (higher) temperature, such as 212°, may be used, in which case the amount of heat involved is slightly less.

Should the cooling be conducted in a vessel inclosing a greater volume than that occupied by the liquid water, only a part of the vapor will condense, the rest remaining to fill the surplus space at the existing temperature. This vapor will of course have associated with it its latent heat of vaporization, and therefore the heat value found will be less than that obtained when all the water is condensed.

(e) Another calorific quantity, known as the "**lower heat value,**" is used by engineers but is not very accurately defined. It is generally assumed to be the heat obtained if all the water formed remains saturated or superheated vapor at the temperature of the products of combustion. This would be numerically

less than the higher heat value already given, by an amount equal to the heat above 60° per pound of vapor in the flue gas.

The accurate determination of the heat which could be obtained by cooling and condensing, under constant pressure, the water vapor contained in flue gases is more or less complicated in most cases. It is first necessary to determine the weight of vapor per cubic foot of gas, its partial pressure, and its temperature. From this data its state can be determined either from steam tables, or from a diagram similar to that of Fig. 34 drawn for water vapor. With the state known, the heat which would be liberated per pound during cooling and condensing under constant pressure can be found from the steam tables. The same result can be closely approximated by the use of the following formula,* which gives very closely the heat above 32° F. per pound of water vapor in the air, or in products of combustion,

$$\Delta Q = 1058.7 + 0.455\, t_1 \text{ B.t.u.}, \quad . \quad . \quad . \quad . \quad (364)$$

in which

ΔQ = B.t.u. per pound above 32° F.; and

t_1 = temperature of vapor in products of combustion (= temperature of gas).

If it is assumed that the liquid resulting from condensation could be cooled only to the temperature t_2° F. (say the room temperature of about 60°, instead of to 32°), then,

$$\begin{aligned} \Delta Q_{t_2} &= (1058.7 + 0.455\, t_1) - (t_2 - 32) \\ &= 1090.7 + 0.455\, t_1 - t_2. \quad . \quad . \quad . \quad . \quad (365) \end{aligned}$$

Thus, every pound of water vapor which escapes uncondensed in the products of combustion will carry with it an amount of heat equal to ΔQ_{t_2}, which is, therefore, unavailable for other purposes. Since every pound of hydrogen burns to nine pounds of water, it follows that the lower heat value per pound of available hydrogen is

$$L.H.V. = 62{,}000 - 9\,(1090.7 + 0.455\, t_1 - t_2). \quad . \quad (366)$$

This expression shows that the lower heat value is really a variable, depending for its value on the lowest temperature t_1 attained by the products of combustion before leaving the apparatus which they are supposed to heat, and also on the tem-

* For explanation of this formula and further details see "Experimental Engineering," Carpenter and Diederichs, p. 467.

perature t_2, which is generally assumed as about 60°. For a value of t_1 equal to 1500° F., and t_2 equal to 60°, the difference between the lower and higher heat values is about 15,000 B.t.u.; for t_1 equal to 500° F. the difference equals about 11,000 B.t.u.; and for t_1 and t_2 both equal to 60° F. the difference is still about 9500.

(f) The value ordinarily used for engineering purposes, and which may therefore be called the **"engineering lower heat value,"** is generally taken at 52,000, which corresponds to a value of t_1 equal to about 530° F. with $t_2 = 60°$ F. It is evident that this may be merely a very rough approximation in many cases.

When this value is used, the lower heat value of the hydrogen in a fuel which contains H pounds of that element and O pounds of oxygen is

$$B.t.u. = 52{,}000\,(H - O/8). \quad . \quad . \quad . \quad . \quad (367)$$

(g) In some cases it is more convenient to use *heat values per cubic foot* of hydrogen rather than per pound. These can easily be obtained by dividing the values already given by the specific volume of hydrogen. There will obviously be as many different values as there are temperature and pressure combinations; hence the conditions under which the cubic foot of gas is to be measured should always be specified. At a temperature of 32° F. and under a pressure of 14.7 pounds per square inch the specific volume of hydrogen is 178 cubic feet. Therefore, the heat values per cubic foot under these conditions are

$$\text{Higher heat value} = \frac{62{,}000}{178} = 348 \text{ B.t.u.} \quad . \quad . \quad (368)$$

and

$$\text{Lower heat value} = \frac{52{,}000}{178} = 292 \text{ B.t.u.} \quad . \quad . \quad (369)$$

(h) In this connection it should be noted that although the heat value per pound of hydrogen is considerably higher than the heat value per pound of carbon monoxide, the values per cubic foot of material are more nearly equal. Thus the value per cubic foot of CO at 32° F. and at atmospheric pressure is about 335 B.t.u., which is but slightly less than the upper value for hydrogen and is considerably greater than the lower value.

This relation is of particular importance in engineering, because:

(1) There are a large number of commercial gases containing both hydrogen and carbon monoxide and it is possible to regulate the relative proportions of the two to a certain extent.

(2) It is generally the volume of the gas which is to be handled, and not its weight, which determines the dimensions and cost of apparatus and cost of operation; and

(3) Under most engineering conditions it is the lower heat value of hydrogen, not the higher, that is made available.

241. Hydrocarbons. (a) Combustibles composed of hydrogen and carbon in combination are known as "hydrocarbons." There are many kinds which differ as to the relative proportions of H and C contained. They burn to form the ultimate products CO_2 and H_2O, but the process is often very complicated. The exact combustion behavior of all the common hydrocarbons is not yet well known but experiment shows that in many cases a number of reactions go on before the actual combustion process is completed.

(b) It is very common practice to assume that when a hydrocarbon containing C pounds of carbon and H pounds of hydrogen is burned it should liberate

$$(C \times 14{,}600) + (H \times 62{,}000) \text{ B.t.u.}, \quad . \quad . \quad (370)$$

but such calculations seldom check with the actual values. This is explained in part by the fact that the hydrocarbon is already a chemical compound and must be broken up to enable the individual elements to combine with oxygen. When this occurs a quantity of heat must be absorbed or liberated, thus diminishing or increasing the amount liberated during the formation of CO_2 and H_2O. Many empirical formulas have been developed to take account of such effects, but none of them are entirely satisfactory for all cases.

(c) In many instances the approximate calculation by Eq. (370) is sufficiently exact, but when great accuracy is desired a determination should be made by using a "Fuel Calorimeter," which will be briefly described in Section 244.

The experimentally determined and calculated calorific values of several of the principal hydrocarbons are given in Table XXIII.

TABLE XXIII. — CALORIFIC VALUES OF HYDROCARBONS.

Name.	Molecular Formula.	Weight in Lbs./Cu. Ft., Atm. Pres. and 32° F.	Calorific Value Experimentally Determined, B.t.u./Lb.		Calorific Value Calculated, B.t.u./Lb.
			Higher.	Lower.	Higher.
Methane....	CH_4	0.04464	23,842	21,385	26,455
Ethane.....	C_2H_6	0.08329	22,399	20,434	24,080
Ethylene...	C_2H_4	0.07809	21,429	20,025	21,370
Acetylene...	C_2H_2	0.07251	21,429	20,673	18,240

242. Combustion of Sulphur. Sulphur burned in oxygen forms sulphur dioxide. The reaction is given by the equation

$$S + O_2 = SO_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (371)$$

The weights combined are

$$32 \text{ of } S + (2 \times 16) \text{ of } O = 64 \text{ of } SO_2,$$

and dividing by 32 gives

$$1 \text{ of } S + 1 \text{ of } O = 2 \text{ of } SO_2. \quad . \quad . \quad . \quad . \quad . \quad (372)$$

Then, for each pound of sulphur, one pound of oxygen is needed for complete combustion and 2 pounds of SO_2 result. To furnish the pound of oxygen approximately 4.35 pounds of air are required. The reaction is accompanied by the liberation of about 4000 B.t.u. per pound of sulphur.

243. Combustion of a Mixture of Elements. (a) If the symbols represent the pounds of each of the respective elements present in a *mixture*, and if it is supposed that the oxygen present is already in combination with hydrogen, then, from the preceding paragraphs, it is evident that for complete combustion there are needed

$$\text{Pounds of Oxygen} = 2\tfrac{2}{3}\, C + 8\,(H - O/8) + S; \quad (373)$$

to furnish this would require 4.35 times as much air, or

$$\text{Pounds of Air} = 11.6\, C + 34.8\,(H - O/8) + 4.35\, S; \quad (374)$$

and the volume of this air at a temperature of 62° F. and at atmospheric pressure is found by multiplying by the specific volume 13.14 (from Table XX), giving

$$\text{Cubic Feet of Air} = 153\, C + 454\,(H - O/8) + 57\, S. \quad (375)$$

The volumes at other temperatures and pressures can be found from the relation $(PV/T)_1 = (PV/T)_2$.

(b) The heat liberated when such mixtures are burned can be conveniently determined by the use of what are known as *Dulong's formulas*. These are:

$$\text{Higher B.t.u.} = 14{,}600\,C + 62{,}000\,(H - O/8) + 4000\,S. \quad (376)$$

$$\text{Lower B.t.u.} = 14{,}600\,C + 52{,}000\,(H - O/8) + 4000\,S. \quad (377)$$

It will be noted that these formulas are merely the summations of the heat values given before for the individual elements.

As already explained, if there are chemical combinations which must be broken up, the heat associated with the separation must be considered besides that given by Dulong's formula. Thus Eqs. (376) and (377) do not apply to hydrocarbons, although their use will give the approximate heat values.

244. Fuel Calorimeters and Heat Value. (a) In the absence of satisfactory methods of calculating the heat liberated during combustion, the scientist and the engineer have developed instruments, known as *Fuel Calorimeters*, for measuring the energy as liberated.

Practically all of them operate in the following way: A known weight, or volume, of the combustible is burned within the instrument under such conditions as to insure as nearly as possible complete combustion, and the heat liberated is absorbed by water or similar liquid in an enveloping jacket. By measuring the temperature rise of the liquid, and correcting for radiation loss from the instrument, the heat liberated is obtained; and from the known weight of the material burned, the heat which would be liberated per unit weight may be calculated. This value is known as the Heat Value, Calorific Value, or Heat of Combustion of the material. In engineering work it is generally expressed in B.t.u. per pound of material, or, in the case of gases,* per cubic foot at standard conditions.

(b) In all calorimeters the jacket temperature is near that of the room and the products of combustion are cooled to approxi-

* It is almost standard practice to use weight as the basis for solid fuels and volume as the basis for gases. For liquids both weight and volume are used, though weight is probably given the preference. Whenever there is a possibility of confusion the unit should be given in the statement of results. If a cubic foot of gas " at standard " is used, the so-called standard should be defined.

mately the temperature of the jacket before leaving the instrument; thus the heat measured is generally assumed to be that obtained by bringing the products of combustion down to the initial temperature of the combustible material. This is seldom really accomplished and an error from this source is therefore introduced into practically all commercial calorific determinations. This method gives what is commercially called the "higher heat value," when the combustible contains available hydrogen.

It has already been pointed out that even with the temperature of the combustion products reduced to 60° F. there may be a considerable discrepancy between the values thus obtained and the true higher value.

An accurate method of stating calorific values would be to give the heat liberated when material at 32° F., or 60° F., is burned and the products are cooled to the original temperature, allowance being made in each case for humidity in the products of combustion. In the present state of the art, however, such refinements are not warranted.

245. Flue Gas Analysis. (a) In connection with tests of furnaces, boilers, and similar apparatus in which fuel is burned, it is often necessary to analyze the flue gases in order that certain efficiencies and losses can be calculated and that the conditions of combustion can be determined. In these analyses the quantities of gases present are generally expressed in "volume percentages." For example, gases *a*, *b*, and *c* may be said to constitute respectively 10 per cent, 20 per cent, and 70 per cent of the total volume (= 100 per cent) resulting from their mixture.

(b) In making the analysis, a measured volume of the mixed gases at atmospheric temperature and pressure is successively brought into contact with appropriate reagents, each one of which absorbs but one constituent gas; then, by noting the corresponding decreases in volume under atmospheric conditions, the volume-percentages of the various constituent gases can readily be determined.

(c) To give a clearer idea of this process, assume that the cubical vessel shown in Fig. 319 (*a*) incloses a volume of 100 units, that it is filled with a mixture of gases, say CO_2, CO and N, and that the pressure of the mixture is atmospheric. Each of

the constituents evidently occupies the entire volume, that is, each is evenly distributed throughout the vessel; each exerts a definite "partial pressure"; and the sum of these partial pressures equals the atmospheric pressure. If it were possible to collect each of the constituents and isolate it from the others by flexible diaphragms, as shown in Fig. 319 (*b*), and if each of the constituents were decreased in volume until its pressure became equal to atmospheric, then the sum of all the volumes would equal the original volume, provided the temperature remained the same. That this is true will be shown in the following paragraphs.

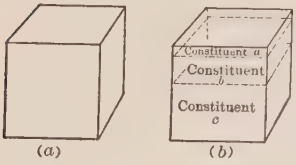


Fig. 319.

(d) Assume, for instance, that in Fig. 319 (*a*) the partial pressures are aP_a, bP_a, and cP_a, a, b, and c being fractions and P_a being atmospheric pressure. If the total pressure of the mixture is atmospheric, the sum of the three partial pressures must equal P_a, that is

$$aP_a + bP_a + cP_a = P_a, \text{ or } a + b + c = 1.$$

If the volume of the vessel is V, then the constituent with partial pressure aP_a will have to be given a volume

$$V_1 = (V \times aP_a) \div P_a = aV,$$

in order to raise it to atmospheric pressure when isolated; the constituent with partial pressure bP_a will have to be given a volume $V_2 = bV$; and the remaining constituent, a volume $V_3 = cV$.

From the relations between a, b, and c it is obvious that

$$V_1 + V_2 + V_3 = V \ (= 100 \text{ by assumption}).$$

(e) Thus a, b, and c not only represent the partial pressure fractions, but also give the fractions of the total volume that would be occupied by the constituents when reduced to the volumes they would have when isolated and raised to atmospheric pressure without change of temperature. As was stated, the so-called percentage by volume of any constituent is therefore merely the percentage of the original volume of the mixture which that constituent would occupy if existing alone at the

same pressure as that exerted by the mixture and at the same temperature.

(f) It is important to note in connection with flue gas analyses that the water vapor content is never determined in ordinary engineering apparatus. The greater part of this water is condensed and disappears by mixing with water contained in the apparatus used in making the analysis. The gaseous mixture is, however, practically always saturated with water vapor during the entire analysis, and although this water exerts a partial pressure, this latter affects each of the constituents proportionately, hence its influence is really negligible. Though water vapor is present the results on the *percentage* basis are therefore the same as those for dry gas.

246. Weight of Flue Gases. (a) In many engineering computations it is necessary to determine the weights of the gases resulting from the combustion of a given fuel under given conditions. Such calculations are simple when one knows (1) the analysis of the flue gases, (2) the analysis of the fuel, and (3) the moisture content of the air.

(b) As was seen in the preceding section, the volumetric analysis of the flue gas is the equivalent of isolating the constituent gases and reducing them to the same pressure and temperature. Then from Avogadro's hypothesis it follows that the number of molecules of each of the gases present must be directly proportional to the volumes (V) which the gases occupy, hence the products (mV) of these volumes by the respective molecular weights (m) of the gases, give measures of the relative proportions by *weight* of the gases present in the mixture; the sum (ΣmV) of these products gives a measure of the weight of the whole mixture; and the weight percentage of any constituent is evidently

$$\text{Per cent weight} = mV \div \Sigma mV. \quad . \quad . \quad . \quad (378)$$

Thus, if the mixture is composed of CO_2, O, CO, N, H, and SO_2, and if the relative volumetric proportions of the constituents are represented by their chemical symbols, then the *equivalent molecular weight* of a unit volume of the mixture is

$$\Sigma mV = (44\ CO_2 + 32\ O + 28\ CO + 28\ N + 2\ H + 64\ SO_2), \quad (379)$$

and the weight percentage of CO_2 for example is found by dividing 44 CO_2 by Eq. (379).

(c) It is generally most convenient to express the constituents in terms of their weight *per pound of carbon burned.* The weight corresponding to the mV value of the carbon represented in Eq. (379) is evidently

$$\textit{Weight of } C = 12\,(CO_2 + CO); \quad . \quad . \quad . \quad (380)$$

hence the weight of any constituent *per pound of carbon* is found by dividing its mV value by $12\,(CO_2 + CO)$. Thus, the weight of nitrogen is

$$w_N = 28\,N \div 12\,(CO_2 + CO);\ \textit{per lb. of } C \quad . \quad . \quad (381)$$

the weight of free hydrogen is

$$w_H = 2\,H \div 12\,(CO_2 + CO); \quad . \quad . \quad . \quad (381a)$$

and similarly for the other constituents.

The total weight (w) of *dry* gas mixture per pound of carbon actually burned is given by dividing Eq. (379) by Eq. (380). When simplified this becomes

$$w = \frac{11\,CO_2 + 8\,O + 7\,(CO + N) + H'2 + 16\,SO_2}{3\,(CO_2 + CO)}, * \quad (382)$$

in which, as before, the symbols represent the relative volumes of the gases they symbolize.

(d) To find the total weight of "wet" gases per pound of carbon, it is necessary to add three more items:

(1) The weight of water, in the fuel per pound of carbon, as found by analysis;

(2) The weight of water carried by the air supplied for combustion, per pound of carbon, which can be found from psychrometric observations; and

(3) The weight of water formed by the hydrogen burned, per pound of carbon. This can be found as follows, when the fuel analysis is known: Let w_H' be the *weight* of *free* hydrogen, *per pound of carbon* in the fuel, and let H, as before, be the *volume* of hydrogen not burned (per pound of C), as found in the analysis of the flue gases; then $w_H' - \{2\,H \div 12\,(CO_2 + CO)\}$ is the weight of hydrogen burned, and the resulting weight of water is

$$\text{Weight of } H_2O = 9\,\{w_H' - [2\,H \div 12\,(CO_2 + CO)]\} \quad (383)$$

* If the gases CH_4 and C_nH_m are present, this expression should have $4\,CH_4 + 7\,C_nH_m$ added to the numerator, and the parenthesis in the denominator should include $CH_4 + 2\,C_nH_m$, it being assumed that the C_nH_m is all Ethylene (C_2H_4).

247. Percentage of Excess Air. It is now possible to derive perfectly general expressions for the percentage of excess air and for the excess coefficient, — that is, expressions which are not limited, as are Eqs. (352) to (355), to the case of the combustion of carbon alone.

If the symbols represent relative *volumes* as before, then, according to Eq. (381) the total weight of nitrogen in the flue gases per pound of carbon is $28\,N \div 12(CO_2 + CO)$, or $7\,N \div 3(CO_2 + CO)$; hence, the oxygen which accompanied this nitrogen must be

$$\text{Total oxygen} = \frac{23}{77} \times \frac{7\,N}{3\,(CO_2 + CO)} \text{ pounds.} \quad . \quad (384)$$

The weight of oxygen not used is, similarly,

$$32\,O \div 12\,(CO_2 + CO) = 8\,O \div 3\,(CO_2 + CO),$$

where O is its volume, which is assumed to be known from the volumetric analysis of the flue gas. But part of this unused oxygen could have been utilized had combustion been perfect. Thus a weight equal to $\frac{16}{28} \times \frac{28\,CO}{12\,(CO_2 + CO)} = \frac{4\,CO}{3\,(CO_2 + CO)}$ might have been used for burning the CO to CO_2; and a weight equal to $8 \times \frac{2\,H}{12\,(CO_2 + CO)} = \frac{4\,H}{3\,(CO_2 + CO)}$ might have been used for burning the free hydrogen in the flue gas. The true weight of excess oxygen per pound C is, therefore,

$$\text{Excess } O = [8\,O - 4\,(CO + H)] \div [3\,(CO_2 + CO)] \quad (385)$$

Subtracting this from the total oxygen (Eq. (384)) gives the weight of required oxygen, per pound of C as

$$\text{Required oxygen} = \frac{\frac{23}{77} \times 7\,N - [8\,O - 4\,(CO + H)]}{3\,(CO_2 + CO)}\,^{*} \quad . \quad (386)$$

Then, since the *percentage of excess air*, x, is equal to excess air (or O) divided by the total required air (or O), it follows that

$$x = \left\{ \frac{8\,O - 4\,(CO + H)}{\frac{23}{77} \times 7\,N - [8\,O - 4\,(CO + H)]} \right\} \times 100.\dagger \quad . \quad (387)$$

* If the gases CH_4 and C_nH_m are present, $4\,CH_4 + 7\,C_nH_m$ should be added to the parenthesis in the numerator and $CH_4 + 2\,C_nH_m$ should be included in that in the denominator. This neglects any N in the fuel.

† To account for the gases mentioned in the two preceding footnotes, $4\,CH_4 + 7\,C_nH_m$ should be added in the parentheses in the numerator and denominator.

The *excess coefficient*, X, is therefore

$$X = 1 + \frac{x}{100} = 1 + \left\{ \frac{8\,O - 4\,(CO + H)}{\frac{23}{77} \times 7\,N - [8\,O - 4\,(CO + H)]} \right\}. \quad * \quad (388)$$

248. Stack Losses. (a) In connection with tests of furnaces, boilers and similar apparatus it is customary to determine the amount of heat carried away by the gases passing up the stack, or as it is often called the " heat lost in flue gases."

(b) Before taking up the complete method of calculating these losses, a simplified theoretical discussion will be considered in order to bring out certain fundamental relations. For this purpose the **case of dry carbon only**, burned with dry air, will be analyzed. In connection therewith the combustion may occur under any one of three sets of conditions, as follows:

Case 1. Complete combustion with theoretical air supply;

Case 2. Complete combustion with excess air; and

Case 3. Incomplete combustion with deficiency of air.

It is obvious that Case 1 is merely a limiting value between Cases 2 and 3, and hence need not be considered separately.

(c) *With Excess Air* (Case 2) the only loss to the stack under the assumed conditions is that due to sensible heat of the CO_2, the nitrogen, and the excess air in the flue gases.

It was shown in Sect. 238 (f), that with x per cent excess air the so-called "products of combustion" resulting from the complete burning of *one pound of carbon* would consist of 3.67 pounds of CO_2, 8.9 pounds of N and $0.116\,x$ pounds of air; hence the total weight of flue gas, per pound of carbon, is given by the equation,

$$\text{Pounds of flue gas} = 3.67 + 8.9 + 0.116\,x \quad . \quad . \quad (389)$$

As this waste gas leaves at a temperature considerably higher than that at which the constituents entered the furnace, it carries with it sensible heat which should have been used. With weight determined the corresponding loss of heat can readily be computed when the specific heat and the temperature of the flue gas are known.

As flue gases of boiler furnaces generally leave with a temperature less than 700° F., it is customary to neglect the variations of the specific heats with temperature, and as the specific heat of the flue gas is nearly the same as that of air, it is also customary

* See last footnote on page 497.

to neglect the change with variations in x. The specific heats assumed for the mixture by different writers generally fall between 0.22 and 0.24; with average excess coefficients, 0.24 is a satisfactory figure. With this assumption, the approximate formula for loss of heat in the flue gas, per pound of carbon burned, for Case 2 is

$$B.t.u.\ loss = 0.24\,(3.67 + 8.9 + 0.116\,x)\,(t_f - t_a), \quad (390)$$

where t_f is the temperature of the flue gas and t_a is the atmospheric temperature. Since each pound of carbon should liberate 14,600 B.t.u. the per cent loss of heat is given by

$$Per\ cent\ loss = \frac{0.24\,(3.67 + 8.9 + 0.116\,x) \times (t_f - t_a)}{14{,}600} \times 100. \quad (391)$$

Values obtained from this equation are plotted in the upper, right-hand quadrant of Fig. 320 and the resulting curves serve to show how the stack loss varies with different values of x and of $(t_f - t_a)$. Actually, because of increases in the specific heats with temperature and excess air, the losses would increase somewhat more rapidly than these curves show.

(d) *With Deficiency of Air* (Case 3 above) there are two stack losses to be considered — that due to sensible heat, and that due to the heat value of the *CO* (or of the *CO* and *C*) not burned.

The weights of gas (per pound *C*) present with y per cent deficiency of air will be given by (from (k) of Sect. 238)

$$Pounds\ of\ Flue\ Gas = 3.67\left(1 - \frac{2\,y}{100}\right) of\ CO_2 + 2.33\,\frac{2\,y}{100}\ of\ CO + 8.9\left(1 - \frac{y}{100}\right) of\ N. \quad (392)$$

Assuming the specific heat 0.24, this would give, per pound of carbon, a stack loss due to *sensible heat* of

$$(B.t.u.)_S = 0.24\left[3.67\left(1 - \frac{2\,y}{100}\right) + 2.33\,\frac{2\,y}{100} + 8.9\left(1 - \frac{y}{100}\right)\right](t_f - t_a). \quad (393)$$

Since each pound of *CO* could give 4300 B.t.u. if burned, there is also, per pound of *C*, *a loss due to the CO* equal to

$$(B.t.u.)_{CO} = 2.33 \times \frac{2\,y}{100} \times 4300 \quad (394)$$

provided the deficiency (y) is not greater than 50 per cent.

The total loss with deficiency of air less than 50 per cent is evidently equal to $(B.t.u.)_S + (B.t.u.)_{CO}$. This has been plotted in the lower right-hand quadrant of Fig. 320 for different temperatures of gas.

If y exceeds 50 per cent some of the carbon is not burned at all and the losses would therefore be still greater. However, as this is a case not ordinarily approached in practice it need not be considered here.

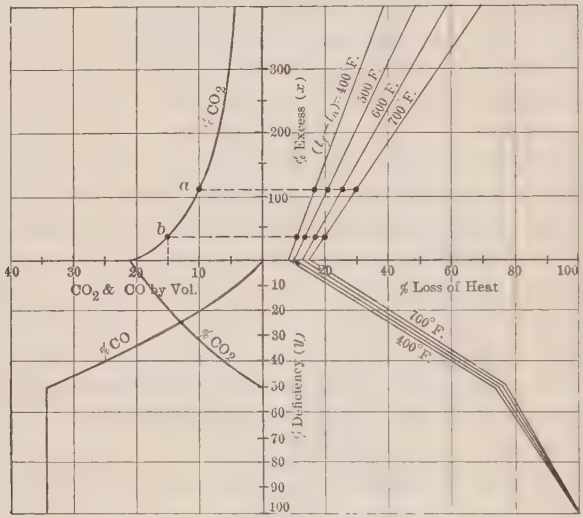


Fig. 320.

The losses resulting from a deficiency of air are shown in the lower right-hand quadrant of Fig. 320.

(e) The completed chart of Fig. 320 includes the curves previously given in Fig. 316, and serves to show in a general way how the losses vary with different temperatures, different quantities of air and different flue gas analyses. It must be borne in mind that certain broad assumptions were made to simplify the derivation of this chart and it is therefore only approximately correct. From an inspection of the curves it at once becomes apparent that losses due to excess air are much less than those due to deficiencies, for example, with flue gas at 500 degrees, the loss occasioned by 100 per cent excess air is equalled by that due to only about 8 per cent deficiency.

In using the chart, however, it is important to note that comparisons of losses incident to using different percentages of excess air should not necessarily be made on the basis of the same temperature — for, ordinarily, larger amounts of air bring about a reduction in temperature of the flue gas.

(f) The foregoing applies to the combustion of carbon alone. In the **actual case** the " flue gas " usually contains CO_2, CO, N, O, H, hydrocarbons and water vapor and therefore differs somewhat from the case just considered. The stack losses in the actual case may be conveniently divided into three distinct parts:

(1) That part represented by the *sensible heat* of the *dry flue gas*, not including the moisture that may be present;

(2) That part due to *incomplete combustion* of some of the constituents of the fuel — this includes the potential heat of the unburned C (in the smoke), CO, H and hydrocarbons;

(3) That part represented by the latent and sensible heat of the water vapor (*moisture*) in the flue gas.

The methods of determining each of these losses will now be considered.

(g) The weight (w) of dry flue gas (per pound of carbon) in the fuel can be found by Eq. (382) in Sect. 246 and it is common practice to use 0.24 for the $\overline{C}_p$ of the mixture. Hence the approximate loss in the *sensible heat* in the dry flue gas (per pound of carbon burned) is

$$(B.t.u.)_S = 0.24\, w\, (t_f - t_a) \quad . \quad . \quad . \quad . \quad . \quad (395)$$

(h) A more accurate method of finding this loss is to first determine (as in Sect. 246) the weight (w_n), per pound of C, of each of the constituent gases, get its mean specific heat $\overline{C}_{p_n}$ from Fig. 318 for the temperature range, then compute the sensible heat it carries away; and finally take the summation for all the constituents. Thus the total sensible heat carried away by the dry flue gases is (per pound of carbon burned)

$$(B.t.u.)_S = \Sigma\, (w\overline{C}_p)_n \times (t_f - t_a) \quad . \quad . \quad . \quad . \quad (396)$$

(i) The stack loss due to *incomplete combustion* is (per pound of C)*

$$(B.t.u.)_I = C \times 14{,}600 + CO \times 4300 + H\,(52{,}183 - 4.095\, t_f + 9\, t_a), \quad (397)$$

* Neglecting hydrocarbons.

in which the symbols represent weights (per pound of C) of the respective substances, and the parenthetical quantity is obtained by subtracting from 62,000 (which is the higher heat value of H) the value $9\ (1090.7 + 0.455\ t_f - t_a)$ as previously given in Eq. (366).

(j) The heat loss due to the *moisture* in the flue gas depends on the source of this water vapor. That moisture which is humidity in the air used for combustion is already vapor and merely becomes superheated in the furnace; hence the heat it carries away is, per pound of C,

$$(B.t.u.)_A = A \times C_p\,(t_f - t_a), \quad . \quad . \quad . \quad . \quad . \quad (398)$$

where A is the weight of moisture in the air used per pound of C.

The loss of heat due to the moisture (M pounds per pound of C) originally in the coal and due to the water formed by the combustion of hydrogen (M' pounds per pound C) is, from Eq. (366),

$$(B.t.u._M = (M + M')\ (1090.7 + 0.455\ t_f - t_a) \quad . \quad . \quad (399)$$

per pound of C.

The total loss of heat per pound of carbon in the fuel is therefore, $(B.t.u.)_S + (B.t.u.)_I + (B.t.u.)_A + (B.t.u.)_M$.

CHAPTER XXIX.

ACTUAL COMBUSTION OF FUELS—FURNACES AND STOKERS—OIL BURNERS.

249. Introductory. In a preceding chapter the physics and chemistry of combustion were discussed for *theoretical* cases only. The study of the *actual* process of combustion in furnaces, which will now be taken up, is more complicated because of the wide variation in composition of the fuels and because there is a great diversity of conditions under which the combustion takes place. In fact, there are so many variables involved that it is substantially true that in no two distinct cases does combustion occur under identical conditions; and even in the same furnace the conditions are constantly varying. It is, therefore, impossible to give detailed discussion of all the possible cases which might occur. There are, however, certain broad general principles, which, if understood, will be of great value in the solution of problems of combustion which arise in actual furnace operation and these will be brought out in the discussion which follows.

250. Air Supply. (a) In the actual case, as in the theoretical one, it is essential that there be furnished a proper amount of air to supply the oxygen needed for combustion. The exact quantity necessary depends on the composition of the fuel and can readily be computed by the method given in Sect. 243, if the chemical analysis is known. The approximate amount of air required is often determined, however, by assuming that the *combustible part* of the fuel is *pure carbon*, each pound of which requires 11.6 pounds of air for complete oxidation and results in 12.6 pounds of flue gas. But, for most practical purposes it is sufficiently accurate, and is on the side of liberality, to assume the entire weight of coal to be composed of carbon, and then use these same values as per pound of coal. It should be noted, however, that the richer the fuel is in combustible hydrogen the

greater will be the proportion of air needed, since one pound of hydrogen requires 34.6 pounds of air or about three times as much as is needed per pound of *C*.

(**b**) In the ideal case, with fuel containing only carbon, each per cent deficiency of air has been seen to result in 1.4 per cent loss of heat because of incomplete combustion (Sect. 236 (g)). As the same thing is substantially true in the actual case, great care must be exercised to insure an adequate supply of air at all points in the fuel bed. As the bed usually varies in thickness and in compactness and texture, the air will meet with less resistance in passing through certain portions than through others. Hence to insure against a deficiency at any point, it is necessary to furnish an amount of air somewhat in excess of what would theoretically be required if it were uniformly distributed and properly mixed with the combustible material. *Excess air* is not without its disadvantages, however, as it dilutes the furnace gases and lowers their temperature, which results in a decrease in the boiler efficiency. Although its presence is thus detrimental, it is much less so under ordinary conditions than is a deficiency, as was made clear in Fig. 320. Hence excess air should always be present but in as small amount as is consistent with satisfactory combustion. Usually an excess coefficient *X* of from 1.5 to 2 times the theoretical amount, on a basis of carbon, is used, i.e., from 18 to 24 pounds of air per pound of combustible. And, as before, it is usually sufficiently accurate to assume the whole of the coal to be carbon, and to use these values as per pound of coal. Experience shows that if less than 1.3 times the theoretical quantity is used, the amount of *CO* formed is generally prohibitive even if the greatest care is exercised in operating the furnace.* But even if the air supply is adequate it does not follow that the combustion is complete, as will be seen in the next section.

(**c**) With pure carbon as the fuel and with the theoretical air supply, there would be about 21 per cent by volume of CO_2 in the flue gas, as was shown in Sect. 238 (**c**). Excess air will result in a decrease of the volume per cent of CO_2 in the manner shown by

* Even when considerable excess air is furnished there may be some *CO* formed in the thicker and more compact portions of the fuel bed because of local deficiency of air. Further, flue gas analyses may also show *CO* which was formed by processes which will not be discussed until later.

the curve in Fig. 321. As the combustible part of the coal is mostly carbon these same percentages hold substantially in the actual case.* Thus, a knowledge of the CO_2 content in the flue gas indicates in a general way the operating conditions within the furnace and enables the boiler attendant to intelligently adjust the air supply.

Experience has shown that if the supply of excess air is such as to give CO_2 by volume between 10 per cent and 15 per cent, the furnace will be operating at its highest efficiency, the exact best percentage varying with different conditions. A value below 10 per cent nearly always indicates too great an amount of air and a value above 15 per cent is generally indicative of too small an amount, as it is usually accompanied by the formation of prohibitive quantities of *CO*. In Fig. 321 the region for the best results is that shown by the portion of the curve lying between (*a*) and (*b*); in Fig. 320 it falls between the points bearing similar letters; and it approximately corresponds with excess coefficient (*X*) between 1.3 and 2.0, given in (**b**) of this section.

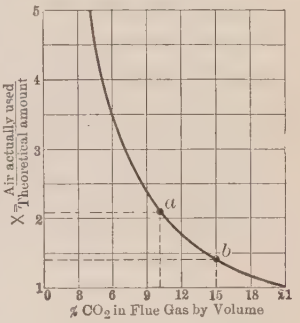


Fig. 321.

(**d**) In order that the boiler attendant may obtain an indication of the amount of air being supplied, various devices known as *CO_2 Recorders*, Econometers, Combustion Recorders, Composimeters, etc.† are used to indicate the CO_2 content of the flue gases. Some of these appliances operate, or indicate, intermittently, some continuously, and some give a continuous graphical record so that the owner or manager of the plant can check the operation over any desired period of time. In the use of all these instruments it is, of course, necessary to obtain samples of gas truly representative of the average and to guard against the infiltration of air through the boiler setting or the flues between the furnace and the sampling point.

* Although the percentage of CO_2 is somewhat less because of the other combustible and noncombustible constituents present in the flue gas in the actual case.

† For description and method of using such apparatus see Carpenter and Diederichs, "Experimental Engineering," published by John Wiley & Sons.

251. Conditions for Complete and Smokeless Combustion. (**a**) If air is passed upward through a deep bed of ignited carbon devoid of volatile matter, there is a tendency for any CO_2 that is formed in lower layers to be reduced to CO when coming into contact with the carbon above. If this CO is not subsequently supplied with a proper *amount of air* while still at a high temperature it will pass off unoxidized and this will result in a loss of heat which would otherwise be made available. It is, therefore, important that an adequate air supply and a suitable *temperature* be maintained in the upper part of, and just above, the bed of fuel. This air may either pass through the bed or be supplied from above.

The foregoing applies of course to the combustion of coke and charcoal as well as to carbon. Anthracite coal, which is mostly fixed carbon, behaves similarly, but in this case there is also a small amount of volatile matter which must be properly burned. These fuels, which have little or no volatile matter, give *short flames* above the fuel bed, the flames being due to the combustion of CO and the small quantity of volatile matter present.

(**b**) When coal possessing a considerable amount of *volatile matter* is placed on a hot bed of fuel, the greater part of the volatile portion distills off as the temperature rises, and the residue, which is coke, burns in the manner just described. The more serious problem that confronts the engineer in this case is the complete oxidation of the combustible part of this volatile matter. Evidently in the ordinary up-draft furnaces that are fired from above the combustion of this part of the fuel must occur above the fuel bed, just as is the case with CO; and in order that the combustible gases may be completely burned, the following four conditions must exist:

(1) There must be *sufficient air* just above the fuel bed, supplied either from above or through the fuel bed itself; (2) this air must be properly distributed and intimately *mixed* with the combustible gases; (3) the mixture must have a *temperature* sufficiently high to cause ignition (some of the combustible gases, when mixed with the burned gases present above the fuel, have an ignition temperature of approximately 1450° F.); and (4) there must be sufficient *time* for the completion of combustion, that is, the combustion must be complete before the gases

become cooled by contact with the relatively cold walls of the boiler (which are at a temperature of about 350 degrees) or with other cooling surface.

(c) To prevent the *stratification* of the air and gases, special means are sometimes adopted, such as employing steam jets above the fire and using baffle walls, arches, and piers in the passage of the flame, to bring about an intimate mixture.

(d) In order that the air used above the fuel bed shall not chill and extinguish the flame, it should be *heated* either by passing it through the fuel bed, or through passages in the hotter parts of the furnace setting, or in some other way before mingling with the gases; or else the mixture of gases and air should be made to pass over or through hot portions of the fuel bed, or should be brought into contact with furnace walls, or other brickwork, which is at a temperature sufficiently high to support the combustion.

(e) In order that the flame shall not be chilled and extinguished by coming in contact with cold objects, it should be protected by the hot furnace walls until combustion is complete. The *furnace* should have proper volume to accommodate the burning gases, and, when the conditions are such that the flame is long, the distance from the fuel bed to the relatively cold boiler surfaces with which the gases first come in contact, should be at least as great as the length that the flame attains when the fire is being forced. The *length of flame* depends on the amount and character of the volatile matter in the fuel, on the rapidity of combustion and on strength of draft. It varies from a few inches, with coke and anthracite coal, to 8 feet or even more with highly volatile coals — even 20 feet has been reached with some western coals.

(f) In order to have complete combustion of *all* the fuel in a furnace it is necessary that uniform conditions prevail throughout the fuel bed; and to bring this about it is essential that the fuel itself be uniform in character. Therefore, the best results are obtained with coal that has been graded as to *size*. Especially is this true with anthracite coal which ignites slowly and is more difficult to keep burning than volatile coals. This coal requires a rather strong draft and unless the bed is uniform the rush of air through the less dense portions tends to deaden the fire in those regions, hence good results can be obtained with this

coal only when it is uniform in size and evenly distributed. The more common sizes of coal are given in Tables XVI and XVII, on pages 465 and 466.

(g) **Smoke** may be composed of unconsumed, condensible tarry vapors, of unburned carbon freed by the splitting of hydrocarbons, of fine noncombustible matter (dust), or of a combination of these. It is an indication of incomplete combustion, and hence of waste, and in certain communities is prohibited by ordinance as a public nuisance. Smoke can be avoided by using a smokeless fuel, such as coke or anthracite coal; or, when the more volatile coals are used, by bringing about complete combustion of the volatile matter. In general, the greater the proportion of the *volatile content* of the coal the more difficult it is to avoid smoke, though much depends on the character of the volatile matter. Coals which smoke badly may give from 3 to 5 per cent lower efficiencies than smokeless varieties.

For each kind of coal and each furnace there is usually a range in the *rate of combustion* within which it is comparatively easy to avoid smoke. At higher rates, owing to the lack of furnace capacity, it becomes increasingly difficult to supply the air, mix it and bring about complete combustion. Hence when there is both a high volatile content in the coal and a rapid rate of combustion it is doubly difficult to obtain complete and smokeless combustion.

However, although smoke is an indication of incomplete and hence inefficient combustion, it may sometimes be more profitable, because of lower price or for other reason, to use a coal with which it is difficult to avoid smoke, provided the latter is not a nuisance or is not prohibited by statute.

252. Value of Coal as Furnace Fuel. (a) The principal factors which determine the *commercial* value of coal used in furnaces are: (1) price per ton, (2) calorific value, (3) moisture, (4) volatile matter, (5) ash, (6) clinkering tendency, (7) sulphur content, (8) skill and attention required in firing, (9) suitability for the furnace and grate in which it is to be used, (10) size of coal and (11) available draft. These will be briefly discussed in this section.

(b) As exposure to weather (sun and rain, humidity, etc.), during transportation and storage, may affect the amount of

moisture and may also alter the chemical composition and heat value of the fuel, especially if rich in volatile matter, the various *analyses* to which the coal may be subjected should be made after the coal is received, or *as received*, if it is desired to determine its value to the *consumer*. The calorific value is preferably determined by using a fuel calorimeter (see Sect. 244); it may, however, be approximated by any of the methods given in Sect. 227(a) to (d). The moisture, fixed carbon, volatile matter, and ash per pound of material may be found by making a proximate analysis (Sect. 226(c)).

(c) If payment is made on the basis of weight of coal "as received," and if the heat value is stated per pound of "dry coal," part of the expenditure is for an unknown weight of moisture and the true value of the coal is unknown. Evidently, from the consumer's standpoint, the purchase *price* should depend directly on the calorific value per pound of the moist coal *as received*. In any case the ultimate test of the commercial value is the cost per B.t.u. *delivered*, or the number of B.t.u. received for a unit of money expended, other things being equal.

(d) The *moisture* in the coal is undesirable as it not only (1) reduces the heat value per pound of material fired, but (2) adds to the transportation expense per B.t.u. delivered, and this in direct proportion to its weight, and (3) decreases the furnace and boiler efficiency since it becomes superheated steam, thereby absorbing heat (latent and sensible), which is carried up the chimney with the flue gases. The heat thus carried away per pound of moisture is the same as that per pound of water vapor formed from the combustion of hydrogen and is given approximately by Eqs. (364) or (365). Roughly, the loss of the total heat value of dry fuel is about $\frac{1}{10}$ per cent for each per cent of moisture present. In eastern coals the moisture normally ranges from 1 to 5 per cent, and in western coals from 3 to 15 per cent.

(e) Coals in which the *volatile matter* is proportionately very high usually give very long flames, and cannot be burned completely or smokelessly unless used with furnaces of proper type, size, and proportions and unless special means are provided for regulating the air supply above the grate. Even with the most careful management it is usually difficult, and in some cases impossible, to obtain complete combustion with such coals even

though an extreme amount of air is used; hence, the calorimetric test is not a true measure of the commercial value of such fuels in furnaces. Fig. 322 shows in a very general way how the efficiency of combustion varies with the percentage of volatile matter in the dry combustible.*

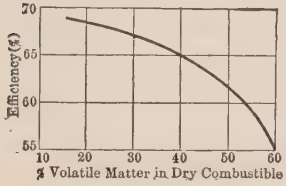


Fig. 322.

Coals *moderately* rich in volatile matter, such as semibituminous and the less volatile bituminous coals, not only have the highest calorific values (as shown by the Mahler curve in Fig. 315), but, when properly fired, generally produce the highest efficiencies of any of the coals used, and with suitable conditions and reasonable attention can be burned smokelessly, or practically so.

(f) The *ash* detracts from the value of coal in a number of ways. The greater its percentage the more difficult it is to obtain complete combustion because of its tendency to pack and obstruct the passage of air; also the greater may be the proportion of coal lost through the grates with the ash, and the less is the capacity of a given furnace because of the reduction of combustible per square foot of grate area. The way in which the value of the fuel decreases as the percentage of ash increases is shown in Fig. 323, which in a general way applies to any kind or grade of coal. When the ash constitutes 40 per cent of coal, the fuel is practically valueless in ordinary furnaces.

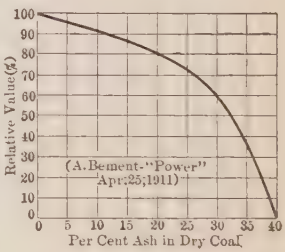


Fig. 323.

The expense of generating a given amount of heat is increased (1) by the cost of transporting the inert matter in the coal, (2) by the transportation and disposal of the ash, (3) by the extra labor involved in handling the larger weight of material, (4) by the unconsumed coal carried through the grates with the ash (which may be from 10 per cent to 60 per cent of the latter), and (5) by the heat absorbed by the ash (specific heat = 0.2 to 0.24) and carried with it to the ash pit. In commercial coals the ash

* "Steaming Tests of Coals," Bull. 23, U. S. Bureau of Mines. Page 233.

generally ranges from 4 per cent to 25 per cent of the total weight.

The smaller the size of coal the more difficult is it to remove the inert portion, hence the greater is the proportion of ash present, as is shown by curve 1 in Fig. 324 for one particular kind of coal.

(g) If the ash is fusible at a comparatively low temperature, it will form *clinkers* when a hot fire is maintained, as when the capacity of the furnace is being forced. This clinker, of course, detracts from the value of the coal. Steam, or water vapor, passed through the fire with the air, is supposed to decrease the tendency to clinker because of absorption of heat and the consequent lowering of the temperature of the ash. For this reason steam-blasts are sometimes used under the grate with clinkering coals and often water is kept in the ash pit to furnish vapor. If clinker is formed, the fused mass must be frequently broken up to permit the free passage of air through the fuel bed to support the combustion.

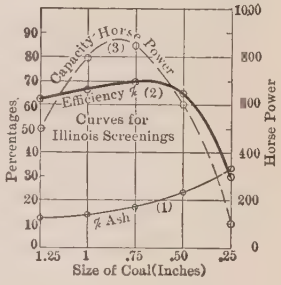


Fig. 324.

(h) *Sulphur* in coal is objectionable not only because of its relatively low heat value, but because of the deleterious effect on the boiler materials, and because it is thought that in some instances it indicates the presence of clinker-forming matter, although clinker also occurs when it is absent. The sulphur should not exceed $3\frac{1}{2}$ per cent.

(i) In general, in using the same coal with a given furnace and draft, the efficiency and capacity of a grate will vary with the *size of the coal*, as is shown by curves 2 and 3 in Fig. 324, for one particular kind of coal tested under a certain boiler.* The best size for given conditions can be determined from experiment or from a study of data relating to similar coals burned under like circumstances. If for some reason it is necessary to burn a given size of a particular coal, there is usually some design of furnace and some set conditions which will give best results; these can be

* Abbott, "Characteristics of Coals," Jour. Western Soc. of Eng'rs., Oct. 16, 1906, p. 528.

determined experimentally if no information on the subject is already available.

In general, the smaller the coal the harder is it to burn completely and the greater is the percentage of unburned coal lost through the grates with the ash. In consequence there is less general demand for the *smaller sizes*, hence they cost less per ton than the larger grades and therefore are widely used in boiler furnaces even though their heat value per pound is low because of the large percentage of ash present. Very fine coal and dust are difficult to burn on ordinary grates as they tend to pack and check the flow of air through the fuel bed, or else, with strong draft, are carried along with the air to be deposited within the boiler setting or to be carried up the stack to become a nuisance to the surrounding neighborhood. They may, however, be burned successfully by the methods which will be given in Sect. 253.

(j) *Caking* of the coal, if excessive, is in general undesirable because of its tendency to prevent the passage of air; but where provision is made to break up the bed, continuously or intermittently, a certain amount of caking may be advantageous.

(k) The different kinds of coal, and the various sizes, do not generally burn at the same rates under equal drafts. With a given grate and draft, it is of course necessary to use a coal which will develop the amount of heat that is needed for the particular purpose for which the furnace is used, for example, if used under a boiler, it must be possible to burn enough coal to evaporate the maximum amount of steam required of the apparatus. Hence under certain conditions the possible *rate of combustion* may have an influence in the selection of a coal. Sometimes when there is uncertainty as to the kind of coal which will eventually be used, — the grate is made of such size that the heat output will be sufficient even though the slowest burning coal is used, — then it will be ample for freer burning kinds — and, subsequently, if desirable, portions of the grate can be blocked off to reduce its area when the latter are used.

Further, there is some rate of combustion (pounds of coal burned per square foot of grate surface per hour) which will give the best combined boiler and furnace efficiency for each kind and size of coal. Fig. 325 shows the variation in the case of one particular kind and size. In general, the rate and heat develop-

ing capacity of the furnace is least with coals low in volatile matter, rich in ash, and small in size, and is, of course, directly dependent on the rate of air supply, that is, on the draft. The best rate to adopt in each particular instance can be determined experimentally, or from a study of similar cases, when data are available.

(l) As the volatile matter is mostly burned beyond the fuel bed, the rate at which coal can be burned *on* a given grate area is largely dependent on the proportion of fixed carbon it contains. The best economies are usually obtained when from 12 to 16 pounds of *fixed* carbon are burned per square foot of grate surface per hour. The *ordinary rates of combustion* (under normal conditions) are about as follows: Anthracite, from 15 to 20 pounds; Semi-bituminous, from 18 to 22 pounds; and Bituminous, from 24 to 32 pounds. Dividing the estimated total weight of coal which is normally to be burned per hour, by the proper normal rate, as here given, results in the necessary grate area and allows for an overload capacity of from 50 per cent to 100 per cent, depending on the intensity of draft that is available. Evidently with anthracite coal there must be a larger grate area for a given total capacity than with bituminous coal.

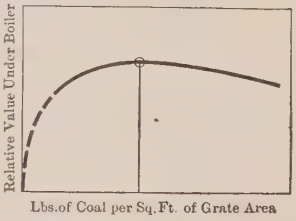


Fig. 325.

Greater rates of combustion are possible; for example, in torpedo boats under forced draft ($4\frac{1}{2}$ inches to 6 inches of water) the rate is from 55 to 65 pounds per square foot; and from 80 to 120 pounds, and even more, have been burned (with air pressure of from 4 inches to 8 inches of water). Rates as high as 90 pounds per square foot per hour are commonly used in locomotive practice where exhaust steam nozzles are employed for inducing strong drafts.

(m) The *maximum capacity* obtainable with a given furnace and with a certain intensity of draft available varies not only with the kind of coal but also with the size. Curve 3, in Fig. 324, shows how it varied with the size of one kind of coal tested under a certain boiler. It is to be noted that the maximum efficiency is not necessarily obtained with the size that gives the greatest capacity.

The capacity per square foot of grate area with *anthracite coal* is limited largely by the fact that if this fuel is burned rapidly, it has a tendency to break up into small pieces which pack and clog the passage of air through the fuel bed. This action also increases the amount of unconsumed coal lost through the grates with the ash and this lowers the efficiency.

253. Burning Powdered Coal. (a) Powdered coal can be burned in much the same way as a liquid fuel (see Sect. (258)) if it is finely pulverized and properly injected into a furnace. When used in this way it has many of the advantages incident to the use of liquid fuel.

However, the cost of crushing and the difficulty of uniform feeding, combined with the complicated apparatus necessary, have thus far prevented any wide use of powdered coal as a boiler fuel although it has been very successfully and widely used for firing cement kilns in regions in which the price of oil is high.

(b) Coal too fine to use on ordinary grates may be *briquetted* by using a suitable binder, as has already been mentioned, and can then be used conveniently and efficiently on ordinary grates.

(c) Where special dumping grates are used with air supplied from below, under pressure that can be readily regulated, very *small anthracite* coal has been successfully burned in the following manner: The fuel bed is not disturbed for cleaning for several hours, the ash being allowed to accumulate; the intensity of air pressure in the ash pit is increased as the fuel bed becomes thicker, but is always such that it becomes atmospheric at the surface of the bed; the products of combustion are carried off by draft induced by a stack or other device above the fuel bed; and the fuel is distributed as evenly as possible in firing. Owing to the fineness of the coal, there is a strong tendency for it to burn out in spots, to prevent which the surface of the fuel bed must be smoothed very frequently by using a distributing bar with a T-head. As the pressure just above the fuel bed is atmospheric there is no objection to the frequent opening of the doors which is necessary for thus working the fire. Culm and other low grade coal, which is ordinarily considered to be waste material, may in some instances be burned with satisfactory results by this method.

254. Selection and Purchase of Coal. (a) The price per ton of the coal delivered is the sum of the cost at the mines and the transportation charges; hence the distance from the mines may have an important bearing on the cost to the consumer. The cost at the mine depends on the difficulty in mining — hence for hard anthracite it is greater than for the softer bituminous coals. Also the price is, of course, dependent on the supply and demand. Thus the smaller sizes, being in the least general demand, are the cheapest per ton at the mine. Grades that are generally considered worthless cost least, hence much attention is being devoted to devising methods for utilizing these grades.

(b) In selecting coal for boilers the problem is to find that kind and size which will give the greatest number of useful heat units, or which will evaporate the largest weight of water, per dollar expended for the fuel and its firing. In default of available information on the subject, a series of tests under varying conditions may be conducted to determine the coal best suited to a given furnace and to find the size of that coal, thickness of fire, strength of draft, method of firing, etc., which will give the best results under the prevailing conditions. But such tests may be as much a determination of the skill of the fireman as of the quality of the coal, and, therefore, may not give the true value that the coal would have when properly used. Although many such tests have been made, the published data of this kind at present available are rather meager.

(c) The principal sources of data are reports of the U. S. Bureau of Mines, of State Geological Surveys, Engineering Experiment Stations, engineering "pocket books," special treatises on fuels, combustion, and boilers, catalogs of boiler manufacturers, etc. Nearly all of these reference books give tables of the chemical and proximate analyses of the fuels from the more important coal fields and while the coals from a given field, and even from the same mine, vary considerably in character, such data may be used in default of tests of samples of the coal actually under consideration.

(d) Some large consumers have adopted the following plan for the selection and purchase of coal: By actual test in their furnaces they determine what kind is cheapest and most desirable to use under the prevailing conditions and thus a standard specification as to heat value, size, ash, volatile matter, mois-

ture, sulphur, etc., is drawn up. Then a "standard price" for such coal is agreed upon with the dealer, with adjustments by premiums and penalties for variations from the specification. The adjustment is directly, or almost directly, according to the variation in the heat value per pound of the moist coal (or coal "as received"), and is dependent on, but not directly proportional to, the variation in the percentage of ash, volatile matter, and sulphur from the standard.

For example, the Interborough Rapid Transit Company (New York City) accepts a run of mine bituminous coal without penalty or premium if it contains 20 per cent or less of volatile matter, 9 per cent or less of ash, and $1\frac{1}{2}$ per cent or less of sulphur. The standard heat value per pound is 14,250 B.t.u. with penalty and premium averaging about one cent per ton per 50 B.t.u. variation from standard. Penalties per ton range up to .18 cents for 4 per cent excess in volatile matter, and to 23 cents for $4\frac{1}{2}$ per cent additional ash, and to 12 cents for 1 per cent excess sulphur.*

Some concerns specify as standard a run of mine, semibituminous coal with 1 per cent moisture, 20 per cent volatile carbon, 7 per cent ash, and not over 1 per cent of sulphur. If x is the per cent of variation from standard, the adjustment in the price is inversely proportional to x for moisture, to $2x$ for volatile carbon, and to $3x$ for ash.

The government awards contracts on competitive bids which are accompanied by specifications of the kind and composition of the coal (ash, B.t.u., and size), which the bidders propose to furnish. The analyses are made on the coal "as received," which takes care of the effect of moisture. The coal is rejected if it clinkers or produces smoke excessively and if it exceeds certain limits in the amount of ash, volatile matter, sulphur, fine coal, and dust. A small variation from the specified standard is tolerated without penalty or premium; but if there is much difference, the price is varied directly with the heat value of the coal "as received" (including moisture) and is adjusted according to a sliding scale for variations in ash and sulphur.†

* Am. El. Ry. Eng'g. Assoc., Report, 1911.

† U. S. Bureau of Mines, Bulletin 11, "Purchase of Coal, etc.;" Bull. 41, "Results of Purchasing Coal under Government Specifications;" and Technical Paper, No. 15, "Sampling Coal, with Specifications."

255. Furnace Operation. (a) The efficiency and capacity obtained with a given coal and furnace depend much on the knowledge, skill, and attention of the furnace attendant and especially is this the case if the furnace is hand fired.

(b) In addition to the considerations already discussed it is important with many kinds of coal to have the right combination of *thickness of fuel bed and draft.* For, in general, with each quality and size of fuel, and with each method of firing and rate of combustion, there is some combination of thickness and draft which will give best results — although there is considerable latitude with some coals. Thin beds tend to let an excessive amount of air pass through and require frequent and careful firing and close regulation of draft. Very thick beds require less attention and give quicker response to sudden increase in demand, but necessitate stronger drafts and are conducive to the formation of *CO.* With a *given draft* and coal the maximum rate of combustion is largely dependent on the thickness of bed. In general, the coarser the coal and the stronger the draft, the thicker should be the bed. But here, again, it seems impossible to give any rule that would be at all general in application; for with anthracite coal, the thicknesses used vary from 4 inches to 12 inches and, with bituminous, from 6 inches to 16 inches, depending on the quality and size of coal, the draft, method of firing, etc. Hence the best combination must ordinarily be found by experience in each instance.

(c) With *anthracite* coal not only must the bed be kept uniform but it must be disturbed as little as possible in cleaning the fire. Hence shaking grates, which cut off the lower part of the bed with minimum disturbance of the upper part, can be used to special advantage with this coal.

(d) The intensity of *draft pressure needed* is directly dependent on the resistance offered to the passage of air through the fuel bed. Its pressure is usually stated in terms of "inches of water." For the usual kinds and sizes of coal and for ordinary conditions the drops in air pressure through the fuel bed are shown by the ordinates of the curves in Fig. 326,* in which the abscissas are rates of combustion expressed as pounds burned per square foot of grate surface per hour.

* Modified from curves given in "Stirling," published by the Stirling Co. (1905).

(e) In an up-draft furnace, fired from above, the air ordinarily enters through the ash doors below the grates, the draft being induced by the stack, which in some instances is assisted by steam blowers, or by fans. The amount of air and the rapidity of combustion can be regulated by adjusting the dampers in the flues leading to the stack, by regulating blowers or fans, and by varying the openings in the ash doors. If coal is fired intermittently, as in hand firing, the layer of fresh coal temporarily

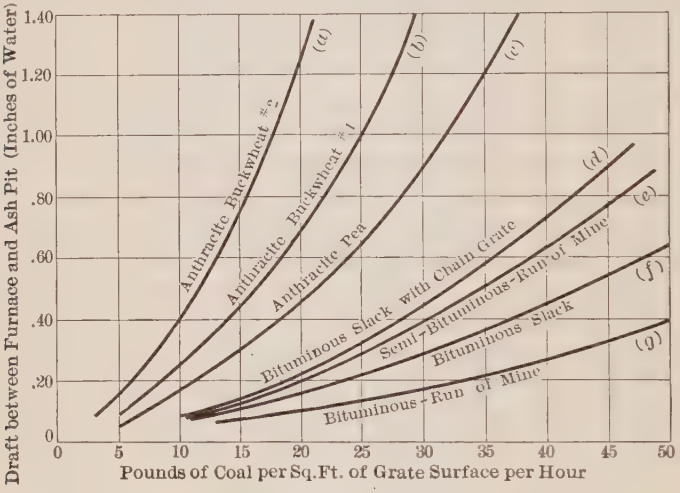


Fig. 326.

chokes the air supply received through the bed and this occurs at the time when the most rapid distillation of volatile matter is in progress. Hence, immediately after firing a fresh quantity of coal, particularly if it is rich in volatile matter, an adequate supply of air should be introduced above the fuel bed and this amount should be diminished gradually as the rate of distillation decreases. This air may be furnished through the fire doors, which may be gradually closed by hand or by some automatic device so arranged that the rapidity of its action can be adjusted to suit the fuel, or it may enter through inlets in the boiler front or in furnace walls, or through passages in the bridge wall at the back of the grate.

In hand firing there is also a loss due to the relatively long period of time during which the doors are open while firing,

which permits large quantities of cold air to enter and pass over the boiler heating surfaces. As the conditions in the furnace vary widely and quite rapidly with the method of firing, the best results can only be obtained by close attention on the part of the attendants and especially is this the case when the fire is being forced — a time when it is the most difficult to give such attention.

(f) The way the coal is distributed on the fuel bed is of importance. In general, there are three *methods of hand firing* commonly used:

(1) In one, called *spread firing*, the fresh coal is each time spread evenly over the entire surface of the bed. This is the method commonly adopted with anthracite coal.

(2) In the second, known as *alternate firing*, fresh coal is placed on but one-half of the grate at a time, which permits excess air to pass through the other, thinner and brighter half for the combustion of freshly evolved volatile matter. *Spot*, or checker, firing is similar; alternate spots on an imaginary checkerboard are fired simultaneously, and, as before, the volatile matter from the fresh coal is supplied with heated air by the excess amount that passes through the remaining portions of the bed. In these methods of firing, the coal is placed each time on the brighter portions of the fuel bed.

(3) In *coking firing*, which is used only with caking coals, the fuel is placed on the front edge of the fuel bed and allowed to coke, the volatile matter passing back over the bed and mixing with the hot air passing through this portion. After distillation is complete the coke is pushed back and distributed over the bed. This method, while effective, does not permit of high rates of combustion.

(g) Evidently the best results can be obtained only when the conditions are maintained uniform, that is, when the coal is fired continuously and uniformly and there is no variation in the air supply. Thus, with hand or *intermittent* firing, the coal should be fired frequently, in small amounts, and it should be distributed with skill, while, in addition, the draft should be carefully adjusted. However, such close attention is opposite to the natural tendencies of furnace attendants, and even with it, it is impossible with some coals to obtain complete and smokeless combustion with hand firing.

(h) By the use of automatic mechanical stokers, however, the coal-feeding operation is made *continuous* and the conditions are kept uniform. They involve but little labor or attention and with them it is possible to obtain practically smokeless combustion with nearly all kinds of coal provided that (1) the grates and the furnace setting are properly proportioned to suit the fuel, (2) proper attention is given to the firing and air supply, and (3) a suitable rate of combustion is used per square foot of grate surface. Mechanical stokers will be described in Sect. 257.

256. Grates and Furnaces. (a) The number of square feet of grate surface may be determined in the manner described in Sect. 252 (1). The width of grate is commonly made equal to the distance between the walls of the boiler setting, and the length of grate is ordinarily found by dividing the desired area by this width. With hand firing, grate lengths up to 10 feet have been used with dumping grates, while with ordinary grates the limit of length is usually 6 feet because of difficulty in cleaning the fire.

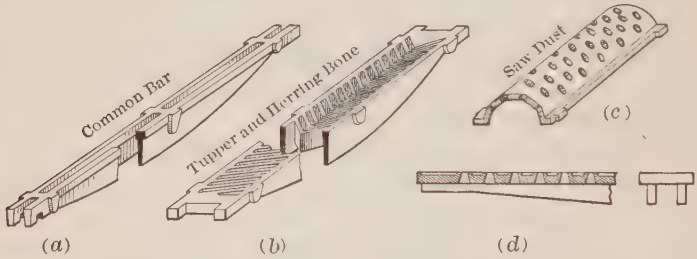


Fig. 327.

(b) To prevent the *grate bars* from burning away they should be of suitable material (white C. I. is generally used), and should be of such shape as to present relatively small surface to the fuel bed and expose large radiating surface to the current of air. They must be in short lengths (not over 3 feet ordinarily), set so as to allow for expansion and contraction and also for warping. They must provide sufficient passage for air, and must permit the ash, but not the coal to pass through. In addition they must be readily cleaned of clinker.

(c) Of the great many *kinds of grate bars* in use four of the most common forms are shown in Fig. 327. In this figure, (*c*)

is for sawdust and the rest are for coal. For fine coal flat plates like (*d*) with small perforations are sometimes used; and these may have the exposed surface recessed so as to become filled with a permanent layer of fine ashes in order to protect the bar from the heat and also to prevent the adherence of clinker to the metal.

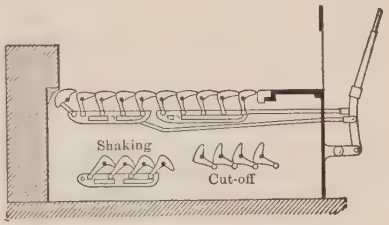


Fig. 328.

(d) Fig. 328 shows one form of *shaking grate*, of which there are a great many different kinds in use. With such grates the fire is not only more easily cleaned, but the fire doors do not have to be opened during the operation; and the bed is disturbed but little, which is especially valuable when anthracite and noncaking coals are burned. Some are provided with means for breaking the clinker, or the caked coal bed, and others for dumping. Their cost is, of course, greater than for ordinary grates but they often give from 1 per cent to 5 per cent better efficiency than flat grates.

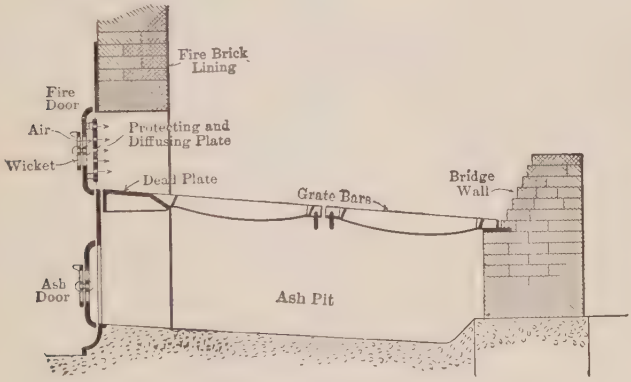


Fig. 329.

(e) Fig. 329 shows a typical *furnace* with fittings. The arrangement shown in Fig. 330 is the *roofless furnace* suitable for coals low in volatile matter, say with less than 20 per cent. Fig. 331 shows the *tile roof arrangement* in which the flame is protected by a roof of refractory material supported by the

lowest row of boiler tubes. This arrangement is suitable for long flame coals, for ordinarily there is little difficulty in making the roof at least as long as the flame.

(f) Fig. 332 shows a *Dutch oven* which can be built in front of any type of boiler. It offers an incandescent roof and walls to

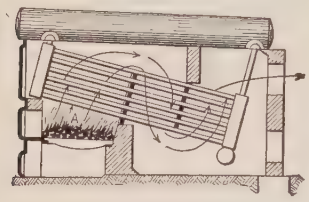


Fig. 330.

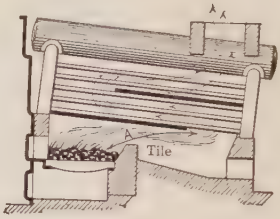


Fig. 331.

reflect the heat and makes possible the complete combustion of volatile matter, but it adds to the radiation losses because of the increased surface exposed. When possible it should be confined within the regular boiler setting, so as to reduce this external surface. In the Dutch oven the roof and walls are sometimes made double with passages between for the circulation of air which is supplied to the furnace. This arrangement serves the double purpose of furnishing hotter air and of reducing the radiation loss by lowering the temperature of the outer walls.

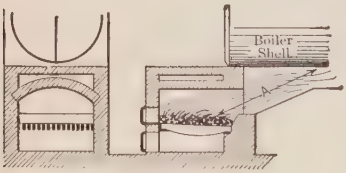


Fig. 332.

(g) In all these figures of furnaces the distance A should at least equal the *flame length*, and the external radiating surface of the setting should be made as small as is expedient. The furnaces and passages for the hot gases must be lined with fire brick, using only the best grades in furnaces in which the firing is forced. Baffle arches, piers, wing walls, etc., are sometimes introduced in the passageway of the gases to mix them and perfect the combustion.

(h) Fig. 333 shows a *down-draft furnace*. In this form of furnace the upper grate bars are cooled and are rather widely spaced. The coal is fired on the upper grate and the volatile matter is carried downward by the draft so as to pass over the hot, partly burned coals which have fallen to the lower grate.

(i) The *efficiency* of a grate is, of course, lowered by the loss of unburned coal with the ashes, while that of the furnace is dependent on the completeness of combustion and on the radiation from the external walls. As these efficiencies have an intimate bearing on the performance of the boiler, they will be considered in connection with the general discussion of boiler efficiencies in Sect. 259.

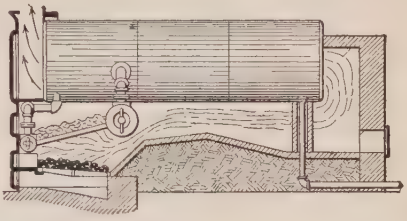


Fig. 333.

257. Automatic Mechanical Stokers. (a) The principal *advantages* derived from the use of mechanical stokers are: (1) Continuous firing and uniform conditions; (2) progressive distillation of volatile matter and proper provision for burning it; (3) a saving of from 30 to 40 per cent labor cost in large plants (in small plants there may be no saving because a certain number of firemen are always necessary and the introduction of stokers may not reduce the number); (4) relief of men from strenuous duties and from exposure to heat; (5) greater ease in obtaining good economy; (6) the elimination to a large extent of the personal element in firing; (7) the greater possibility of smoke prevention with the poorer grades of coal; and (8) greater rates of combustion than are possible without smoke.

(b) The main *disadvantages* (which may, or may not, be present in any given make of stoker) are: (1) Greater first cost (and interest on same); (2) possible lack of durability and greater cost of repairs; (3) cost of power to operate; (4) greater complication; (5) inability to meet sudden changes in load; (6) failure to distribute coal evenly; (7) loss of unburned coal with ashes; and (8) loss due to use of steam in the air-blast. What was said about the design and arrangement of grate bars and furnaces in general in the preceding sections also applies to automatic stokers.

(c) At the lower rates of combustion it is possible to obtain about as good results with hand firing as with automatic stokers, but this involves the employment of painstaking men of great skill who command higher wages than the ordinary. With automatic stokers and furnaces designed to suit the coal and

draft, the best results are obtainable with very little effort or skill on the part of the attendants. Most plants using automatic stokers are also equipped with automatic coal conveying machinery and means for delivering the coal by gravity to the hoppers of the stokers, and in such cases the labor is, of course, reduced to a minimum.

(d) No one type of mechanical stoker is equally valuable for all kinds of coal, but practically any kind of coal can be burned efficiently and smokelessly with a suitable stoker, provided the rate of combustion does not exceed a certain value which is dependent on the kind of coal.*

(e) In hand firing one man can effectively attend to from 200 to 500 boiler horsepower, and at the same time wheel the coal and ashes, and regulate the feed water pumps, draft, etc. In such case from 1000 to 2500 pounds of coal are handled per hour.

When merely firing, with coal delivered by others, one man can hand fire about 1000 boiler horsepower, i.e., handle from 4000 to 5000 pounds of coal per hour.

With automatic stokers provided with coal fed by gravity from overhead bunkers, one attendant can ordinarily care for from 2000 to 4000 boiler horsepower, using from 8000 to 20,000 pounds of coal per hour.

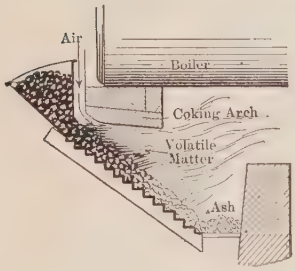


Fig. 334.

(f) Mechanical stokers may in general be classified as:

(a) *Over feed* (including (1) front feed, (2) side feed, and (3) chain grate) and (b) *under feed.*

These will now be discussed in a very general way.

(g) In most *over-feed stokers* (see Fig. 334) the coal is deposited in a hopper from which it is automatically and continuously fed to the grate and made to pass under a more or less extensive *coking arch*, which is maintained at a high temperature and serves the same purpose as the roof of the Dutch oven. Air, heated or otherwise, is usually admitted with the coal under the coking arch. The grate bars are moved in such manner as to carry the bed of coal constantly in one direction and as it progresses it

* Bull. 40, U. S. Bureau of Mines, "Smokeless Combustion," an investigation of several hundred plants.

gradually burns out. As the coal approaches the hotter portion of the fuel bed a *progressive distillation* of the volatile matter occurs. The resulting gas mixes with the air above and the mixture then passes under the coking arch which, being heated to incandescence, reflects the heat from the bright portions of the bed and deflects the gas so as to make it pass over the rest of the bed. Thus the conditions are excellent for the complete combustion of the volatile content.

The rate at which coal is fed from the hopper to the grates and at which it is carried along the latter can be varied and should be so adjusted that combustion of the coal is just completed when the end of the grate is reached. If completed before this point cold air will force its way through the thin bed of ashes at the end and reduce the efficiency; and if not completed unburned coal will be lost with the ashes.

The stokers may be driven in various ways, such as by small steam engines, by electric motors, or by belting from conveniently located line shafting.

(h) Fig. 335 shows diagrammatically a typical arrangement of a *front-feed stoker* with inclined grate. It has a hopper, coal-pusher feeding-device, dead plate, coking arch, and air inlet under the latter. The grate bars are stepped and inclined, and they are mechanically oscillated, or reciprocated, in such way as to cause the bed of coal to gradually descend. The rapidity and amplitude of motion of the pushers and grates can be so regulated that the coal is just burned out by the time it reaches the bottom of the grate. The ashes and clinker become deposited on the ash table, which is dumped by hand from time to time. When the ash table is tilted a guard is brought into position (as is shown at (*a*) in the figure) to keep the fuel bed from sliding down and being dumped at the same time. In the figure, the upper reciprocating grate bars are hollow and the air, which is injected into their interiors by steam jets *S*, issues through openings in the risers of the steps. The lower grates rock or oscillate about the trunnions shown in (*b*) and have replaceable bars. There are, of course, many other designs and arrangements of front-feed stokers.

(i) The typical arrangement of *side-feed* stokers is shown diagrammatically in Fig. 336. Coal is fed into the magazine from above or through doors (*a*) in the front and is pushed, by

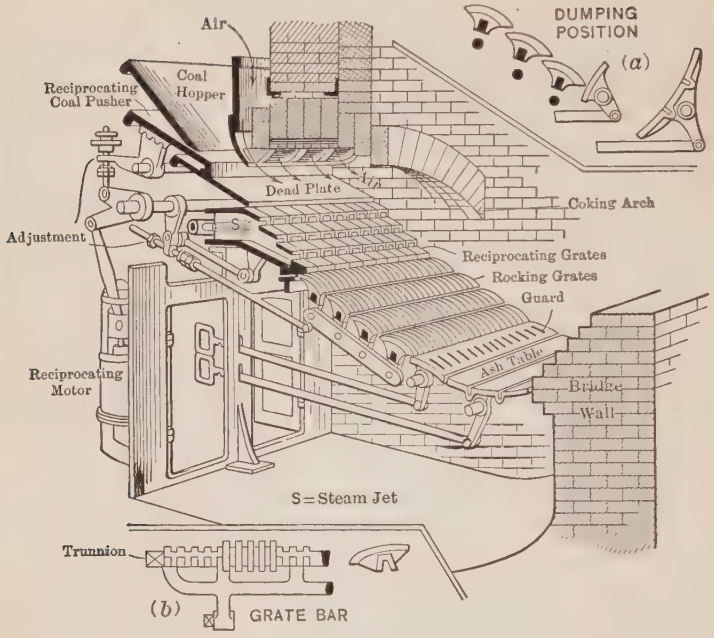


Fig. 335.

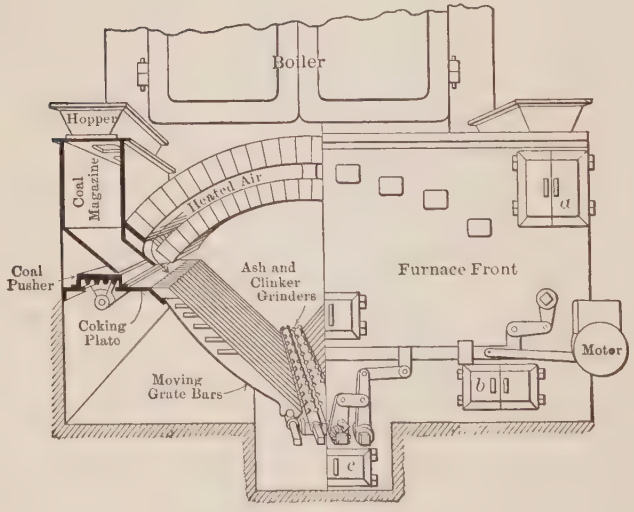


Fig. 336.

some suitable mechanism, to the coking plate at the top of the inclined grates. The whole bed of fuel is covered by a fire-brick arch, and air, which is heated by passing over the arch, is discharged into the furnace just above the entering coal. The coal gradually descends on the inclined grate bars, the alternate ones of which are constantly moving. The ash and clinkers are crushed by rotating (or reciprocating) grinders located at the bottom of the grates. Some grinders are made hollow and are connected to the draft in such way as to cause cold air to pass through them to prevent overheating. When clinkering coal is used, steam (from the stoker engine, if there is one) is discharged through the bed of ashes to reduce the amount of clinker and to make crushing easier.

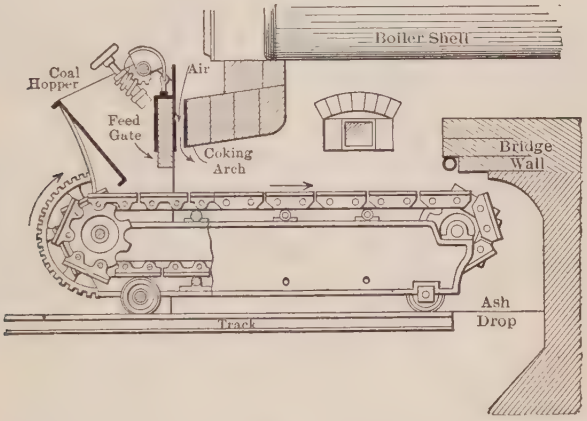


Fig. 337.

The advantageous features of this type of stoker are the large coking spaces, the ample coking arch, and the voluminous combustion chamber. These stokers operate successfully with both uniform and variable loads and under a great variety of conditions. With some, when certain types of coal are used, there is difficulty in getting rid of the ash and clinker. The types differ principally in the manner of feeding the coal and getting rid of the residue.

(j) The typical arrangement of *chain-grate stokers* is shown in Fig. 337. This has the hopper, the coking arch with air ducts

and the feeding device common with the other forms of over-feed stokers already described. The grates consist of a series of endless chains carried on sprocket wheels which slowly rotate and thus carry the coal toward the back of the grate. The whole mechanism is usually mounted on wheels on a track and can be pulled forward for inspection or repairs. These stokers are particularly adapted to the smaller and poorer grades of non-caking coals. To operate satisfactorily the thickness of fire, the draft, and the speed of the grate must be adjusted to suit the load. Combustion should be complete when the coal has just reached the back of the grate.

(k) In the *under-feed* stokers, a typical arrangement of which is shown in Fig. 338, the coal is fed forward from the hopper, by a

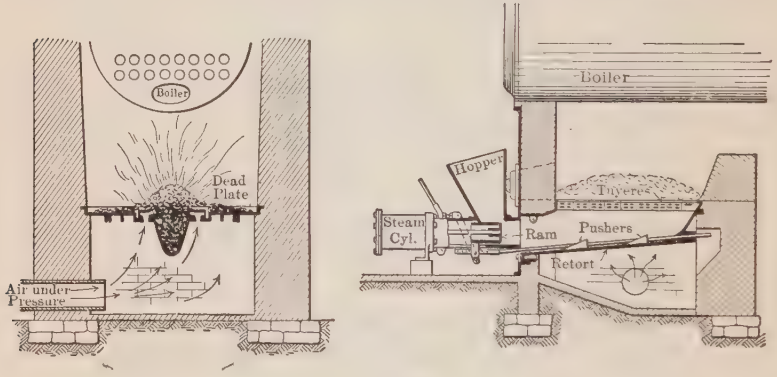


Fig. 338.

reciprocating pusher, as shown (or by a screw conveyer or other suitable feeding device), into a retort, around the upper edges of which are replaceable tuyere blocks, through which air is supplied under pressure. The combustion takes place at the top of the bed towards which the fresh coal is fed from below. The ashes and clinkers fall to the sides of the retort on dead plates from which they can be readily removed through doors in the furnace front. The volatile matter is liberated as the coal becomes heated and this must pass through the intensely hot coals above, where it is mixed with the entering air and is completely burned. Even with volatile coals, the combustion is completed within a very short distance from the surface of the fuel bed, hence only a very short combustion space is necessary. Such stokers give satis-

factory results even when placed in corrugated flues as small as three feet in diameter, such as are used in internally fired boilers.

With such stokers it is necessary to use very strong draft (about three inches of water) which must be furnished by some forced draft system, hence the operation is independent of weather conditions. In some instances the rate at which air and coal are supplied is controlled automatically by the steam pressure. In one such arrangement the speed of the blower-engine is regulated by the steam pressure (a drop in pressure, due to a sudden demand on the boiler, causing an increased speed and hence a greater delivery of air) and the valves for the steam-actuated coal feeder are operated by this same engine; hence the rates at which the air and coal are supplied are changed simultaneously, are kept properly balanced, and the boiler pressure is automatically maintained substantially constant.

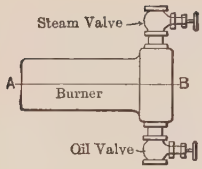
With these stokers it is possible to obtain very high rates of combustion in a very limited space, from 500 to 600 pounds of coal per hour being consumed in each retort. They operate best with bituminous coals which are low in ash and they are not ordinarily satisfactory with fine anthracite coals.*

There are, of course, numerous possible arrangements of such stokers. In some the retorts are inclined and have two horizontal pushers, one above the other.

258. Burning Liquid Fuel. (a) Both crude petroleum and the product of its partial refinement, fuel oil, are very extensively used as fuel in boiler plants. The fuel oil is generally preferable to crude petroleum on the score of safety as, due to the removal of the more volatile constituents during the refining process, its flash point is higher. Most fuel oil also has a lower water content than the crude material and for this reason there is less danger of the flame being extinguished by water collecting in the fuel pipes and then passing as a "slug" through the burner.

(b) To successfully burn fuel oil it is necessary to spray, or "atomize," it very effectively and to mix this in the furnace with the necessary air. The furnace should be well lined with brick which, becoming incandescent during operation, will insure satisfactory combustion so long as there is sufficient air well mixed with the fuel. It is also essential that the furnace be so large and so shaped that the burning fuel does not come in con-

* Bull. 40, U. S. Bureau of Mines.



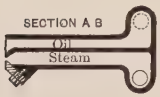


Fig. 339.

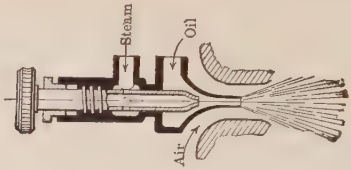


Fig. 340.

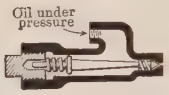


Fig. 341.

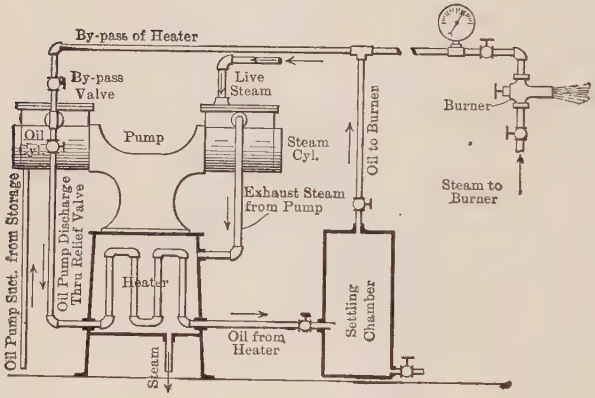


Fig. 342.

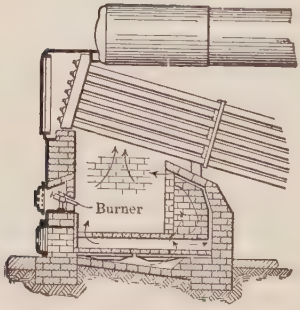


Fig. 343.

tact with boiler heating surface; failure in this respect will result in incomplete combustion of the fuel, as in the case of long flaming bituminous coal, and is also liable to result in the overheating and ultimate failure of the exposed heating surface.

(c) The oil is generally atomized or sprayed by compressed air or by steam, the latter being now the more common method. It is also occasionally atomized mechanically. In most cases the oil is pumped from storage tanks to burners by small steam-driven pumps. On the way to the burners it is heated by means of the exhaust steam from the pumps, after which it enters the nozzles or "burners," from which it is so sprayed as to give a long jet of finely divided fuel which can thoroughly mix with air admitted to the furnace. The amount of steam required in handling the oil varies from about 2.5 per cent to 5 per cent of the total amount generated — usually it is about 3 per cent and is about evenly divided between the pumps and the burners.

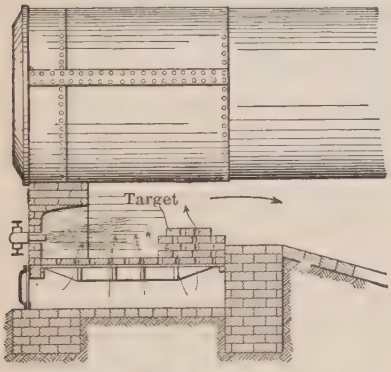


Fig. 344.

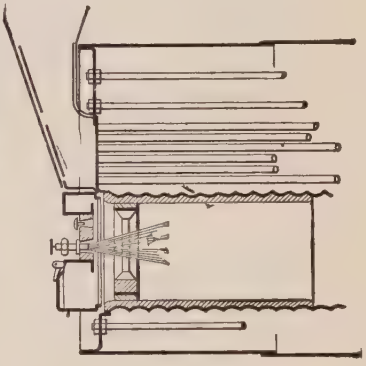


Fig. 345.

(d) The principal *advantages* of burning oil under boilers are:

1. Ease of handling from tank car to furnace, as no manual labor is required even in the smallest plants.
2. Small weight and volume, since the oil has 30 per cent higher calorific value for equal weight, as compared with coal.
3. Lack of clinkers and ash.
4. Higher average operating efficiency because of (*a*) ease of operation, (*b*) ability to properly gauge and maintain necessary air supply, (*c*) smaller excess of air required because of ease of forming good mixture, and

(*d*) more uniform furnace conditions because there is no necessity to open doors at frequent intervals.

5. Practical elimination of soot and smoke.
6. Decreased labor bill in large plants because of ease with which one man can handle several thousand boiler horse power.
7. Ease with which boiler can be made to follow rapid fluctuations of load.

(e) To offset these advantages are high cost of oil fuel in comparison with coal in many parts of the country and the increased danger of fire due to the more inflammable character.

(f) Figs. 339, 340, and 341 show several burners, of which there are a great many other forms in use. One of the numerous possible arrangements of the oil-feeding system is shown in Fig. 342; and in Figs. 343, 344, and 345 are illustrated three of the many arrangements of furnace in use.

258A. Burning Gaseous Fuels. (a) Low-priced, blast-furnace, natural and coke oven gases are frequently used in boilers. Air for combustion is generally admitted through or around the burner, and with gases of high heat value a number of small burners is preferable to a single large one, to prevent a blow-pipe action. The gas and air should have a rotary motion or else a checker-work wall should be used to insure proper mixing. The sizes of gas and air openings depend on the heating value and pressure of the gas. The furnace arrangements are similar to those for oil fuels, with volume from $\frac{3}{4}$ to $1\frac{1}{2}$ cu. ft. per rated horse power. A stack 130 ft. high is ordinarily sufficient.

(b) If a combustible mixture of gas and air is passed through a mass of finely broken refractory material in a tube at such rate that the flame will not "strike back," the combustion will take place at a fixed point in the refractory mass which will become incandescent and transmit heat at rapid rate, by radiation and conduction, to the water surrounding the tube. This method of burning gas, which has been called **Surface Combustion,** has been used in experimental boilers by Bone and others with highly efficient results, due principally to the small amount of excess air, the perfect combustion, and the low temperature of flue gas. The rapid rate of heat transfer permits the use of less heating surface for given output.*

* London Engr., Apr. 14 and Nov. 14, 1911, Jr. A.S.M.E., 1914, Eng. Survey.

CHAPTER XXX.

BOILERS.

259. Losses Connected with Steam Generation. (a) Because of the very intimate connection between the boiler proper * and the other parts which make up the steam generating apparatus, it is most convenient to discuss the losses and efficiencies of boiler, furnace, and grate at the same time. Reference to the energy stream in Fig. 346 will assist in following the discussion.

(b) It should first be observed that it is the function of the furnace to receive fuel, with its supply of heat in latent form, and to make the maximum possible amount of this heat available for use. The furnace may therefore be called the "heat generator." It is then the function of the boiler proper to serve as a "heat absorber" and to transmit to the contained water and steam as large a part of this heat as possible. But losses always occur in making the heat available in the furnace and similarly there are some that are unavoidable in utilizing that heat.

There are in general three *losses in the furnace:* (1) Some of the combustible is not burned, but is lost with the ash; (2) some, which is not so lost, is incompletely burned and passes off with the products of combustion in fine particles; and (3) some of the heat actually made available in the furnace is lost by radiation and cannot therefore be utilized by the boiler.

Thus, only a fraction of the heat originally supplied with the fuel is really brought to the boiler heating surfaces for utilization and part of this must always be unavailable even in an ideal boiler, for after the products of combustion are cooled to the boiler (steam) temperature there can be no further transfer of

* The term "boiler" is ambiguous. It is used to refer to the boiler proper (or vessel containing the water and steam) and also to this element in combination with the furnace, setting and other parts, which collectively comprise the whole steam generating apparatus. However, this should not lead to confusion as the context always makes clear the sense in which the term is used.

heat to the boiler.* This *unpreventable loss* is equal to the heat required to raise the temperature of the flue gases from atmospheric to boiler temperature.

Including this one there are three losses of heat associated with the boiler proper or "heat absorber." These *boiler losses* are: (1) The unpreventable loss (in the ideal case) equal to the heat utilized in raising the flue gases to boiler temperature; (2) a loss resulting from the fact that in commercial boilers the temperature of the flue gases is never reduced to that of the steam; and (3) a loss resulting from the radiation from external surfaces of boiler and setting.

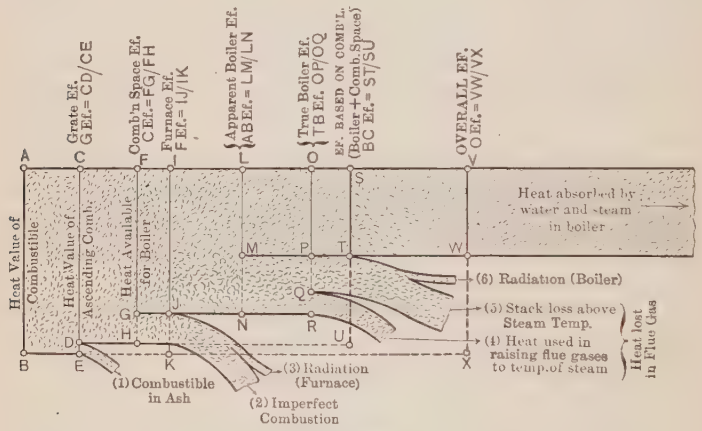

Grate Ef. GEf.=CD/CE
Comb'n Space Ef. CEf.=FG/FH
Furnace Ef. FEf.=IJ/IK
Apparent Boiler Ef. ABEf.=LM/LN
True Boiler Ef. TBEf. OP/OQ
EF. BASED ON COMB'L. (Boiler+Comb. Space) BCEf.=ST/SU
OVERALL EF. OEf.=VW/VX
Heat Value of Combustible
Heat Value of Ascending Comb.
Heat Available for Boiler
Heat absorbed by water and steam in boiler
(6) Radiation (Boiler)
(5) Stack loss above Steam Temp.
(4) Heat used in raising flue gases to temp. of steam
Heat lost in Flue Gas
(1) Combustible in Ash
(2) Imperfect Combustion
(3) Radiation (Furnace)

Fig. 346.

The exact values of the total radiation loss of the complete apparatus and the proportions chargeable separately to furnace and to boiler are generally indeterminate, but they may be approximated more or less closely in some instances.

(c) During a boiler test, it is possible to obtain data which can be used in determining the distribution or destination of the known heat value of the fuel actually fired. The tabulation of such information is called a **heat balance** and accounts for all the heat utilized and lost. It is usually stated both in terms of B.t.u.'s and on the percentage basis. A complete heat balance would include the following eleven items: (1) The heat utilized (absorbed by the water heated and the steam generated); the

* There is an exception to this statement in the special case of boilers that operate on the "counter flow principle." This will be discussed later.

losses due (2) to unconsumed combustible in the ash and (3) to the removal of ash from the ash pit while at high temperature; the stack losses (Sect. 248) occasioned by (4) moisture in the fuel, (5) humidity in the air supplying the oxygen, and (6) water formed by the combustion of hydrogen, and that due to (7) the sensible heat in the flue gas; the losses due to (8) unconsumed *CO*, (9) unburnt hydrogen and hydrocarbons and (10) to the solid fuel (such as fine coal dust and soot) carried off by the draft; and (11) the losses not otherwise accounted for — principally radiation. The sum of the items on the B.t.u. basis must, of course, equal the heat in the coal actually fired, and on the percentage basis it must total 100 per cent.

(d) In real tests it is seldom practicable to make any such complete balance as that just given. It is common practice * to limit it to the following six items: — (1) Heat absorbed by the boiler proper and the losses due to (2) moisture in the coal, (3) moisture formed by the burning of hydrogen, (4) sensible heat in flue gases, (5) unconsumed *CO*, and (6) those not otherwise accounted for (including that due to unconsumed *H* and hydrocarbons, moisture in air, radiation and others not listed above).

A complete discussion of the method of determining the various losses is outside the province of this book. For further details the student is referred to books devoted to Boilers and Furnaces and to Experimental Engineering.

260. Efficiencies Connected with Steam Generation. (a) After the preceding discussion of the losses occurring in boilers and after a study of the energy stream in Fig. 346, it is evident that numerous ratios between the widths of the stream at various points will give efficiencies of the different elements of the steam generating apparatus and of their combinations. The more important of these efficiences will now be given, but as they are clearly shown in Fig. 346 the discussion will be very brief.

(b) Of the combustible placed in the furnace, a part may be lost through the grates with the ash. That which is not thus lost must ascend from the grate as volatile combustible, as gaseous products of combustion, as unburnt solid matter, or as a mixture of these; it will be called " combustible ascending

* A.S.M.E. Power Test Code, page 54.

from the grate " or " ascending combustible." Obviously, the *Grate Efficiency* is

$$GEf = \frac{Weight\ (or\ heat\ value)\ of\ ascending\ combustible}{Weight\ (or\ heat\ value)\ of\ combustible\ fired}, \quad (399x)$$

which is shown in Fig. 346 by the ratio CD/CE.

(c) *The Efficiency of the Combustion Space* (including the coking arch, gas mixing structures, and other parts of the furnace above the grate) is

$$CEf = \frac{Heat\ made\ available\ for\ absorption\ by\ boiler}{Heat\ in\ ascending\ combustible}. \quad (400)$$

This is shown in Fig. 346 by the ratio FG/FH.

(d) The furnace, or " heat generator " includes both the grates and the combustion space. Hence the *Furnace Efficiency* is

$$FEf = \frac{Heat\ made\ available\ for\ absorption\ by\ boiler}{Heat\ value\ of\ combustible\ fired}. \quad (401)$$

$$= GEf \times CEf. \quad (401a)$$

In Fig. 346, FEf is the ratio IJ/IK.

The numerator in Eq. 401 is evidently equal to the sum of (1) the heat absorbed by water and steam, (2) the heat in flue gases leaving boiler and (3) the radiation from the boiler and its walls. Items (1) and (2) can be determined without difficulty, but the radiation losses can in general only be approximated. For this reason the Furnace Efficiency is often omitted from reports of tests.

(e) It has been seen that the heat used in raising the flue gas from the temperature of the atmosphere to that of the steam is not ordinarily available for use in the boiler proper; * hence, if the products of combustion are at a temperature equal to, or below, that of ebullition there will be no heat used by the boiler, even though the furnace itself has high efficiency, — and as far as the boiler proper is concerned, all of the heat is then wasted. To have the boiler use the maximum amount of heat evolved, the unavailable portion must of course be made as small as possible. This useless amount is not only dependent on the temperature difference between the air and steam, but also on the weight of the gas heated. It can, therefore, be minimized

* See footnote on page 534 for exception.

by decreasing the weight of excess air supplied for combustion. Furthermore, the benefit of such reduction is twofold, for not only does it decrease the *amount* of heat unavailable, but it results in higher temperature of the products of combustion, which makes the unavailable portion a smaller *percentage* of the total heat evolved, which in turn increases the efficiency of the steam generating apparatus as a whole.*

(f) The *Apparent Efficiency of Boiler* (alone) may be defined as

$$ABEf = \frac{\textit{Heat absorbed by water and steam}}{\textit{Heat developed in furnace}} \quad . \quad . \quad (402)$$

and in Fig. 346 $ABEf = LM/LN$. The determination of this efficiency involves a knowledge of the furnace losses, — hence, like the Furnace Efficiency, it is difficult to determine accurately.

(g) But some of the heat developed in the furnace has been shown to be unavailable for the ordinary boiler,* and it is hardly just to charge against such a boiler the non-utilization of this portion; hence, the apparent efficiency is not a true measure of the performance in such case. Calling the heat with temperature above that of the steam "potential heat," then, what may be termed the *True Boiler Efficiency* is, evidently,

$$TBEf = \frac{\textit{Heat absorbed by water and steam}}{\textit{Potential heat}} \quad . \quad . \quad (403)$$

In Fig. 346, $TBEf = OP/OQ$, the unavailable heat being shown by QR.

(h) What is called "*Efficiency Based on Combustible*" in the A.S.M.E. Power Test Code (1915) applies to the combined efficiency of boiler proper and combustion space and is expressed as follows:

$$BCEf = \frac{\textit{Heat absorbed by water and steam}}{\textit{Heat available in ascending combustible}} \quad (404)$$

$$= ABEf \times CEf. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (404a)$$

* It has been suggested (Bull. 23, U.S. Bureau of Mines) that the numerators in Eqs. (400) and (401) should include only the heat above the steam temperature. However, while this limitation would be satisfactory for comparison between boilers of the ordinary type, it would be inapplicable to those using the "counter flow" principle. Hence, in this text, the numerator will be taken as the total heat evolved in the furnace, regardless of its temperature.

In Fig. 346 $BCEf = ST/SU$. This efficiency measures the perfection of operation of the boiler and combustion space combined (not including the grate) and as it can be readily determined it is generally given in reports of boiler tests.

"Efficiencies Based on Combustible" have been obtained as high as 85 per cent with oil fuel in short tests under exceptional conditions. In continuous running, 75 per cent efficiency with coal, and 80 per cent with oil, are attainable under uniform conditions. With variable loads and ordinary conditions, average efficiencies of 60 to 65 per cent throughout the year represent good performance.

(i) The *Overall Efficiency* ($OEf = VW/VX$ in Fig. 346) includes the Grate Efficiency and the "Efficiency Based on Combustible," and in the A.S.M.E. Code is termed "Efficiency of Boiler, Furnace and Grate." It is a measure of the perfection of the combined performance of the boiler, furnace, and grate, and is affected by the skill of the firemen, the suitability of the coal and draft, the dropping of coal through grate bars, etc. Hence

$$OEf = \frac{\textit{Heat absorbed by water and steam}}{\textit{Heat in the combustible fired}} \quad . \quad . \quad . \quad (405)$$

$$= BCEf \times GEf = GEf \times CEf \times ABEf. \quad . \quad . \quad (405a)$$

The OEf can be readily determined and hence is usually incorporated in reports of boiler tests. With solid fuels its numerical value is slightly less than the $BCEf$.

(j) Except in the case of "Efficiency Based on Combustible" ($BCEf$) and of Overall Efficiency (OEf), there is lack of agreement among engineers as to the definitions and names of the efficiencies of the various elements of the steam generating apparatus. Hence, before proceeding with the discussions including the use of such terms it is always important to first arrive at an understanding of their meanings. The terms and definitions used in the foregoing treatment appear to the authors to be the most satisfactory ones.

261. Boiler Heating Surface and Heat Transmission. (a) The water heating surface (H.S.) of a boiler is the surface of those parts of the shell which are in contact with water on one side and with the furnace gases on the other. As the transmission of heat from the flue gases to the boiler shell is less rapid

than that from the shell to the water, the heating surface should theoretically be measured on the *gas* side of the plates or tubes. In the case of tubes, however, it is common practice to consider the outer (larger) surface as heating surface, regardless of whether it is exposed to water or gases.

(b) With a given amount of potential heat in hot gases, the more extensive the heating surface the nearer will the flue gases be cooled to boiler temperature and, neglecting radiation, the higher will be the *true* efficiency of the boiler. This is shown by curve E in Fig. 347 where ordinates are efficiencies, or relative performance, and abscissas are extent of H.S. With infinite surface all the potential heat would be absorbed and thus this efficiency would be 100 per cent on this assumption. However, it is, of course, necessary to include the effect of the radiation losses which evidently depend directly on the extent of the radiating surface which is proportional to the heating surface. In Fig. 347 the percentage of this loss is represented by the ordinates of the line R. The net result, or percentage of heat usefully utilized, is given by the difference between the ordinates of the two curves and is shown by line $E-R$. Evidently the maximum efficiency, considering radiation, occurs when the boiler heating surface has an extent represented by the abscissa OM.

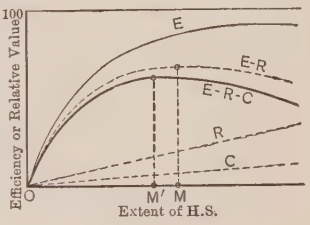


Fig. 347.

(c) Since the cost of boiler, together with that of its floor space and housing, increases with the extent of heating surface, and since the interest on first cost plus the amount set aside yearly for depreciation, insurance and taxes also increases at the same rate, there is also a commercial reduction in value with the extent of surface, which may be shown by some line such as C in the figure. Hence, the true or *commercial value of the heating surfaces* would be shown by some such curve as $E-R-C$, and the maximum value corresponds to a heating surface shown by OM'. Either greater or smaller amounts of heating surface would give less return per dollar expended, hence the extent of heating surface should correspond to this abscissa.

(d) The mean rate of evaporation per square foot of heating surface per hour for the whole boiler is obtained by dividing the

total weight of equivalent evaporation per hour by the total heating surface. From data obtained from tests of many boilers operated at different rates, points may be plotted with abscissas representing these mean rates of evaporation and with ordinates representing either efficiencies or Units of Evaporation per pound of combustible per hour. Average curves drawn with respect to such points resemble those shown in Fig. 348 * and are seen to be similar to *E–R* in Fig. 347. They indicate that the *maximum efficiency* occurs when the "equivalent" mean rate of evaporation for the whole boiler is between 2 and 4 pounds per square foot per hour, corresponding approximately to a transmission of from 1900 to 4000 B.t.u. per square foot per hour.

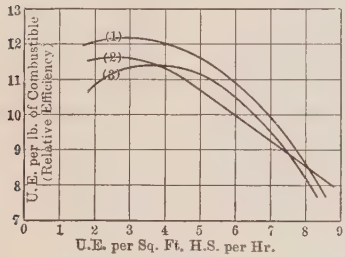


Fig. 348.

(e) But all parts of the heating surface are not equally effective. Evidently those parts in the direct path of the gases are of greater value than those exposed merely to stagnant gases, and those nearest the source of heat are the most effective of any. Heating surface exposed to the "radiant" heat of the fuel bed and burning gases is very much more effective than that not so exposed. Thus, in some cases, the small heating surface immediately over the fire may transmit as much as two-thirds of the total heat absorbed by the boiler, and at this point from 20 to 35 or more pounds of water may be evaporated per square foot of heating surface per hour, whereas the average for the whole boiler may not be more than one-tenth as much. It follows that surfaces farthest away from the furnace must necessarily transmit very much less than the average. Hence importance should be placed not only on the amount of heating surface but also on its distribution and location.

(f) Without going into a detailed discussion of *Heat Transmission* at this point (for this will be given in Chapter XXXV) it will be advantageous to mention here the manner in which the heat generated in the furnace is transmitted to the steam.

Briefly, the heat from the fuel bed is first brought to the heat-

* Such curves are given in Kent's "Steam Boiler Economy" and in Donkin's "Steam Boiler Performance."

ing surface by direct radiation from the glowing coal and burning gases (i.e., as "*radiant*" heat), and by *convection* by the gases which come from the furnace; it is then passed through the metal walls by *conduction*, to be absorbed by the water which may also transport it by convection due to the circulation of this liquid; and finally, when the water has reached the temperature of ebullition, the further addition of this heat results in the formation of the vapor.

(**g**) The *rate of transmission* * per unit of area of heating surface depends, among other things, on (*a*) the difference in temperature between the transmitting and the receiving media; (*b*) the rapidity (velocity) with which the gases are brought in contact with the heating surface, and (*c*) the rapidity with which the heat can be carried away by the water (rapidity of water circulation); (*d*) the amount of scale and grease on the water side of the plate, and (*e*) the amount of soot on the surfaces exposed to the flue gases.

(**h**) The effectiveness of each part of the heating surface is dependent, among other things, on the difference between the temperature (t_g) of the gases on the one side and that (t_w) of the water on the other, i.e., on ($t_g - t_w$). In the case of the ordinary boiler, as has been shown, t_w is constant and equal to the temperature of the steam, since all the water is (approximately) at that temperature. As the gases progress over the heating surface this temperature difference diminishes and the heat transmission per square foot becomes less until the limit of effectiveness is reached, which generally occurs when the temperature difference has been reduced to around 100° to 200° F.

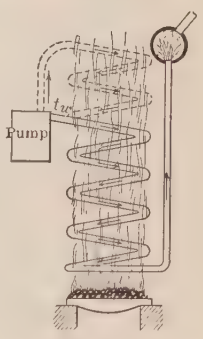


Fig. 349.

(**i**) There is, however, one way of obtaining a value of t_w that is below the steam temperature and therefore of making it possible to absorb more of the heat from the flue gas than can be accomplished in the ordinary boiler; this involves the use of *counter current flow*. The principle under which this operates can be explained in connection with Fig. 349, in which the arrangement is such that the pump

* Reference, U. S. Bureau of Mines, Bull. 18, "The Transmission of Heat into Steam Boilers."

forces the water *downward* through the heating coils, whereas the hot gases pass *upward* — that is, the heat-conveying and heat-absorbing media flow in *opposite directions*. With such arrangement it is obvious that the addition of more heat absorbing coils at the top (as shown dotted) will result in lowering the temperature at which the gases leave, and that by adding a sufficient number this temperature could be reduced to that of the entering water. Hence, with the counter current principle, t_w is not limited to the steam temperature and more heat can be absorbed by the heating surface than is possible in the ordinary arrangement of boiler. Parenthetically it may be remarked that without considerable modification the simple arrangement shown diagrammatically in Fig. 349 would probably not be satisfactory as a boiler element.

The counter current arrangement is approximated in some instances by placing an "economizer" (to be described later) beyond the boiler so that the hot gases after leaving the boiler surrender some of their heat to the water which passes through the economizer on its way to the boiler. So far, the counter current principle has been ignored in the design of most boilers, but it is approximated in a few types.

(j) The rapidity with which the gases flow over the heating surfaces has a twofold influence on the rate of heat transmission: for (1) more heat is conveyed to the surface in a unit of time, and (2) the gases are brought more intimately in contact with those surfaces, since there is less opportunity for a stagnant nonconducting film to adhere to the surfaces.

(k) The rapid *circulation of water* within the boiler is of especial importance when it is necessary to have high rates of heat transmission, for it brings larger amounts of water in contact with the heating surfaces in a given time and also prevents the metal from becoming overheated. This circulation is brought about by providing a free and unrestricted path for the current of water and by applying the more intense heat at the proper point in this path. In Fig. 350, (*a*) and (*b*) show elements of common forms of boilers and the arrows indicate the direction of circulation. The water just above the furnace is less dense than that in the other portions of the boiler, since it has absorbed more heat and is charged with bubbles of steam, and it therefore rises, being replaced by an equal amount of water which descends at

points in the boiler where it is colder and denser. This is the manner in which the current is established and maintained in nearly all the standard types of boilers, as will be seen in studying the figures in the subsequent sections.

(l) In some cases this circulation can be made so powerful that the water in the ascending column can be discharged at an

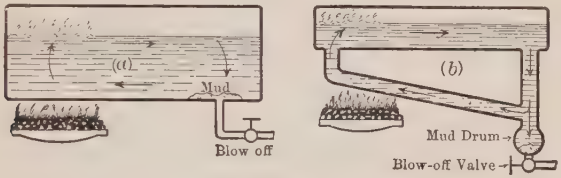


Fig. 350.

elevation even considerably above that of the surface of the body of water from which the descending column receives its supply. This can be accomplished with the arrangement shown in Fig. 351, in which the arrows indicate the direction of flow. The circulation is due to the fact that material in riser *A* is sufficiently charged with vapor to make it weigh less than that in the down-comer *B*, although the altitude *H* is greater than *h*. As the liberation of the steam from the water is supposedly more effective when the ascending column discharges in this manner, some boilers have arrangements somewhat like that shown in the diagram. Sometimes a nonreturn valve like *V* is inserted to insure the proper direction of flow. After the circulation is once established, however, this valve is no longer necessary, as the current is then very positive.

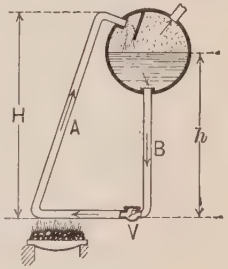


Fig. 351.

(m) The effectiveness of the heat transmission depends on the *cleanliness* of the heating surface. It is diminished by any deposit of soot and dust on the exterior surfaces as well as by any interior coating of soft scale (mud), of hard scale, or of grease.

Water in its natural state contains more or less foreign matter in suspension or in solution. Some of the latter precipitates when the temperature reaches about 200° F., still more when 300° is approached, and the remainder, which is left when the water becomes steam, gradually becomes concentrated until it

reaches the stage where deposition occurs. Deposits on the water side of the walls of the boiler reduce the heat transmitting ability of the plates from 0 to 20 per cent, depending on the thickness of the scale and on the chemical and physical properties of the material.

The formation of scale should be prevented as far as possible by purifying the water before feeding it to the boiler; but even then there will be some deposit formed which must be removed from time to time. Boilers are therefore always so arranged that they can be readily cleaned internally, and so that the deposit shall, as far as possible, occur at points where the heat is the least intense and where the blow-off pipe can be connected (as in Fig. 350) so that the softer material can be removed by blowing off some of the water from time to time. The exterior of the heating surfaces should also be accessible for removing the soot and dust.

262. Boiler Explosions. It has been seen that, by expanding steam, heat-energy can be made available which can be utilized in forcing water and steam through the orifice of a nozzle at very high velocity. As a result of such discharge there is, of course, a force of reaction which will move the nozzle and attached parts unless prevented in some manner. The size of this force depends, among other things, directly on the area of the orifice.

A similar process occurs when a boiler shell is ruptured, for, in passing through the rent in the boiler shell, the steam and boiling water are subject to a decrease from the original pressure to atmospheric, and surrender heat which is converted into the kinetic energy of the issuing mass. The reactive force acting on the boiler shell is dependent in amount on the area of the rent, and may be sufficiently great, compared to the weight of the boiler, to propel the vessel to a considerable distance. In addition to the probable destruction of property and possible endangerment of lives which may result, the escaping steam and water may itself cause considerable damage, — in fact persons near by may be seriously, and perhaps fatally, scalded, even though the reaction is not sufficient to displace the boiler.

With boilers containing little water and having elements which are of small size and so designed as to have small rents when ruptured, the effect of an explosion is less disastrous than

in the case where a large opening can occur and thus instantaneously release a large mass of water and steam.

263. Selection of Boilers. (a) There are a great many items to be considered in the selection of a boiler for a given service; only some of the more important ones can be discussed here. Between the various kinds of boilers which have become well established there is little choice as regards the efficiency, as their performances are substantially equal, hence the selection among such *standardized types* depends largely on general suitability for the conditions of operation and space available, on personal prejudice and familiarity, on convenience in transportation and ease of erection, and on the first cost together with the various other items of expense.

In considering an unfamiliar or untried design the following are some of the items to be checked:

(b) *Suitability.* It should be decided whether or not the boiler is suitable for the coal that is available, and for the kind of grates (or stoker) and furnace best adapted to that fuel. In special cases where the water is bad and the draft poor these items must also be considered. It is not only important that the boiler should have sufficient size to meet the normal demands, but it should have overload capacity sufficient for all emergencies.

(c) *Safety and Durability.* These depend on the design for structural strength, on the character of the materials used (castings under pressure being avoided) and on the character of the workmanship. The arrangement should be such as to avoid stresses due to the unequal expansion and contraction of the different parts of the boiler; and the method of support should be such that the structure, as a whole, is free to adjust itself with change of temperature. There should be no thick plates or other parts (such as boiler joints) and no projecting portions, or plate edges, exposed to the current of the hotter gases; nor should the blow-off pipe be exposed to these gases.

(d) *Accessibility.* The ability to easily reach all parts of the boiler for inspection, cleaning and making repairs, must be investigated. Doors in the boiler setting must be provided for access to all exterior parts; manholes, or handholes, must be so located as to render accessible all internal parts.

In connection with internal *cleaning* cognizance must not

only be taken of the number of manhole and handhole joints to be broken and subsequently made tight, but also of the time required for doing this, for cooling the boiler, and its setting sufficiently to permit of starting such work, and for bringing the boiler into commission again. In some water tube boilers the dust and soot can be blown from the tubes by means of a blast of steam or air issuing from a small pipe which is passed through openings in the front and rear of the boiler or its setting. Other boilers are provided with openings in the side walls for this purpose.

The design of the boiler should be such as to permit of making *repairs* without difficulty. In most types of boilers the principal difficulty is with the tubes. The arrangement should be such as to permit readily of the removal and replacement of any one of the tubes without disturbing the other tubes or other parts. If the tubes are straight but few need be carried in stock, whereas if they differ widely in curvature it may be necessary to have on hand a large collection to meet any emergency that may arise.

(e) *Circulation of Water.* It is necessary to see that the arrangement is such as to allow a free and unrestricted circulation of the water and that the heat is applied at such a point as to establish and maintain the current. The rapidity of circulation is of course limited by the smallest cross-section of the circuit. The arrangement of the structure should be such that there are no pockets where steam can form rapidly and keep the water away from the heating surfaces subject to high temperature, for under such conditions the boiler shell will burn away at such points.

(f) *Circulation of the Furnace Gases.* It is desirable to maintain a uniform velocity of the furnace gases and to avoid sudden contraction and expansion as they proceed through the boiler. Within limits, the greater the velocity the more rapidly will the heat be conveyed to the heating surface and the greater will be the amount of evaporation from a given surface. There should be no pockets where the gas can remain stagnant and it is desirable to have the gas baffled in such a way as to constantly bring the fresher portions into contact with the heating surface as the gas proceeds.

(g) *Dryness of Steam.* To prevent priming, or the entrainment of a considerable portion of moisture in the steam, the liberating *surface* of the water from which steam arises should be ample.

When the water contains certain impurities foaming may occur, and this always increases the amount of entrained moisture. By providing a *large steam space* the life of the particles of steam within the boiler may be made sufficiently long to allow a more or less complete precipitation of the moisture to occur. Provision is often made within the boiler for the separation of moisture by means of "dry pipes," baffles, or other steam separating devices.

(h) *Quantity of Water.* If the boiler contains a large volume of water there is less attention required in maintaining the water level, and the boiler has a greater reserve to meet sudden demands than is the case in boilers having a small volume; but greater damage would ordinarily result in case of explosion. In marine and similar service the greater weight involved is of course objectionable.

(i) *Feed Water.* The boiler feed should be introduced in such manner as not to retard the circulation of the water, and, if cold, should not come in contact with the boiler shell. Certain of the impurities in solution in the entering water precipitate when the higher temperatures are reached and are deposited as mud. The water should be introduced at such a point that this precipitate will be deposited where it will do no damage and from which it can be readily removed, see Fig. 350 (*a*) and (*b*). Sometimes a "mud drum" is provided as in the latter figure, or a "settling chamber," as in Figs. 362 and 363, from which the mud may be blown off from time to time.

(j) *Space Occupied.* In addition to the floor space and height occupied by the boiler and furnace, there must be charged against the apparatus the amount of space that must be provided for the *replacement of tubes* and for cleaning. In some horizontal boilers there must be space in front (or rear) at least equal to the length of the tubes (see Fig. 361). This fixes the minimum distance between parallel rows of boilers or between the boiler end and the wall of the building. In some types of vertical boilers sufficient room must be provided overhead for the replacement of tubes.

When the exterior of the heating surface is accessible for *cleaning from the front* or rear of the setting, the boilers may be arranged in a *continuous* "battery" (with adjacent walls in common), in which case the walls between boilers are thickened slightly. When the cleaning is done *from the side*, the boilers

are arranged in a series of batteries of two each, with sufficient space between the pairs to permit of *access to the openings* in one side of each boiler setting.

(**k**) *Cost.* This, of course, is one of the items of fundamental importance. Besides the first cost of the boiler, with its setting and trimmings, and the expense of transportation and erection, it is necessary to consider charges for up-keep and depreciation. The size of the boiler and furnace and the space necessary for the removal of tubes and for cleaning must also be considered in connection with their influence on the cost of the ground and building.

264. Classification of Boilers. Boilers may be classified in many different ways, only a few of which need be given here.

(**a**) *In Internally fired boilers* the furnace is located within the structure of the boiler and is usually made integral with it, while in *externally fired* boilers the furnace is placed below the boiler proper and is surrounded by a "setting" which is generally of brickwork.

(**b**) In *fire tube boilers* (commonly called "Tubular Boilers") the furnace gases pass *through* the tubes which are surrounded by the water from which the steam is generated; whereas in *water tube boilers* (sometimes called "Tubulous Boilers") the water circulates through the tubes while the hot gases pass over their exteriors. Fire tube boilers are shown in Figs. 352 to 359; and water tube boilers are illustrated in Figs. 360 to 366. These will be discussed later.

(**c**) *Sectional boilers* are composed of small elements so arranged that any rupture which may occur will produce only a relatively small opening and will result in but little damage to the boiler itself and to its surroundings. Such boilers may be shipped in small parts which are assembled when being installed. Examples of this type of boilers are shown in Figs. 360 and 365.

(**d**) In *vertical boilers* the tubes are arranged perpendicularly, or approximately so. In general, such boilers demand less floor space than *horizontal* ones but their height is greater.

(**e**) In *straight tube* boilers it is comparatively easy to clean and inspect the tubes. The use of *curved tubes* is inherent in the design of some boilers and they give a certain degree of flexibility to the structure. (See Fig. 363.)

(**f**) Boilers are also sometimes classed according to their use; for example, there are locomotive boilers, marine boilers, portable

boilers, stationary boilers, etc. The descriptions which will follow, will be limited in most cases to the stationary types.

(g) There are innumerable arrangements of boilers and their settings; only a few of the more typical ones will be considered in the following sections.

265. Internally Fired, Tubular Boilers. (a) Such boilers are generally compact and self contained; they are shipped com-

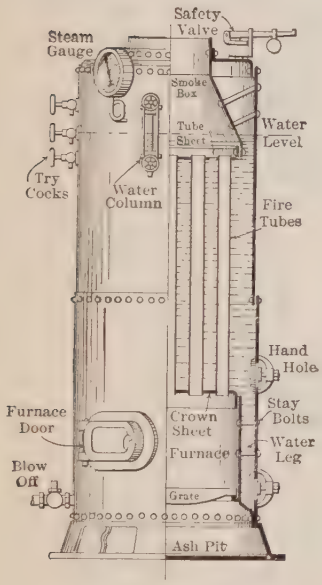


Fig. 352. — Tubular Boiler. Submerged Tube Type.

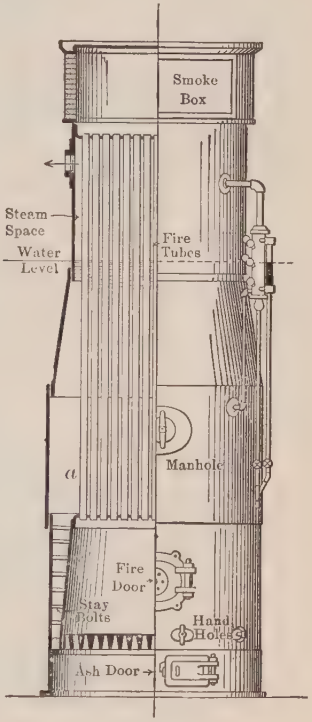


Fig. 353. — Tubular Boiler. Exposed Tube Type.

plete, and immediately upon arrival are ready to connect to the flues and steam system. While they cost more than ordinary boilers, they avoid the expense of special brickwork "setting" and eliminate the possibility of leakage of air through cracks which may develop in such brickwork. Sometimes there is difficulty in transporting the larger sizes.

(b) Fig. 352 shows a small vertical boiler of this kind with

water level above the tubes. Such boilers are of the *submerged tube type*. In Fig. 353 is a somewhat similar boiler in which the tubes extend above the water level — the exposed portions presenting surface to the steam. Boilers of this kind are called *exposed tube* boilers. The one shown in this figure is of such large

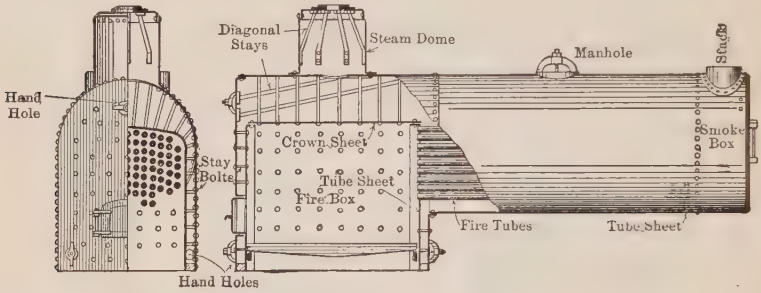


Fig. 354. — Locomotive Type of Boiler.

size that the space at *a* can be occupied by a man while cleaning the tubes, the crown sheet and the plates around the furnace.

(c) Fig. 354 shows a *Locomotive type* of boiler with a steam dome which provides additional steam space. Such boilers are

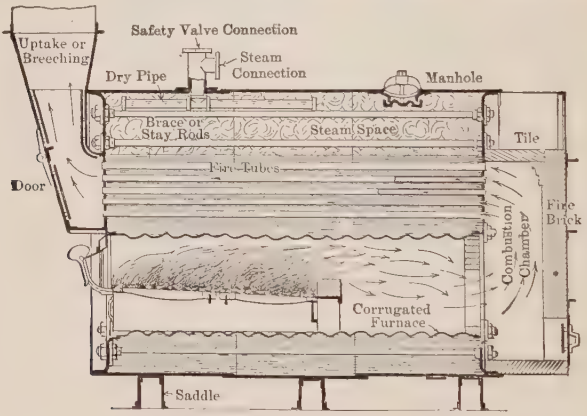


Fig. 355. — Continental Type of Boiler.

not only used for locomotives and for traction engines, but also for stationary service.

(d) In Fig. 355 is a longitudinal section of a boiler of the *Continental type*, the exterior of which resembles Fig. 356. The furnace wall is a cylindrical flue with strengthening cor-

rugations. The combustion chamber is lined with fire brick or other refractory material and is located in a casing of thin metal extending from the main shell of the boiler. These boilers have large liberating surface, voluminous steam space and large volume of water. They usually have either one or two furnace flues, and because they are compact and have short tubes, they can be used in places where the space is limited.

(e) *The Scotch Marine type* of boiler is shown in Fig. 356 and is similar to the Continental except that its combustion chamber

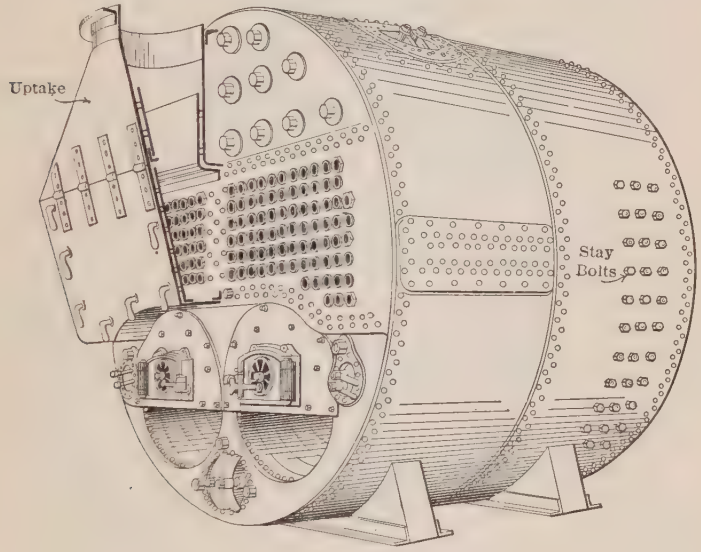


Fig. 356. — Scotch Marine Type of Boiler.

(see Fig. 357) has metal walls and is entirely surrounded by water. As these walls tend to collapse under the external pressure to which they are subjected, they are carefully stayed. Such boilers have from one to four corrugated furnace flues, and their outer shells range from $5\frac{1}{2}$ feet to 16 feet in diameter. Because of the very short tubes, large steaming capacity for space occupied, absence of brick setting, and accessibility, they are particularly adapted to marine service.

266. Externally Fired Tubular Boilers. (a) Boilers of this type generally require a separately constructed "setting" (usually of brickwork with lining of firebrick) to surround the

furnace and boiler. This is so arranged as to properly confine the flue gases and guide them to and from the boiler. It takes considerable time to construct and dry out the brickwork setting

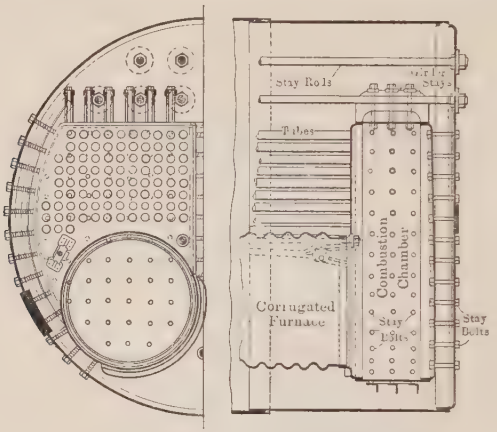


Fig. 357. — Submerged Combustion Chamber.

and the expense involved must be added to the cost of the boiler itself. Such boilers usually occupy more space than internally fired boilers and the setting should be kept in repair so as to

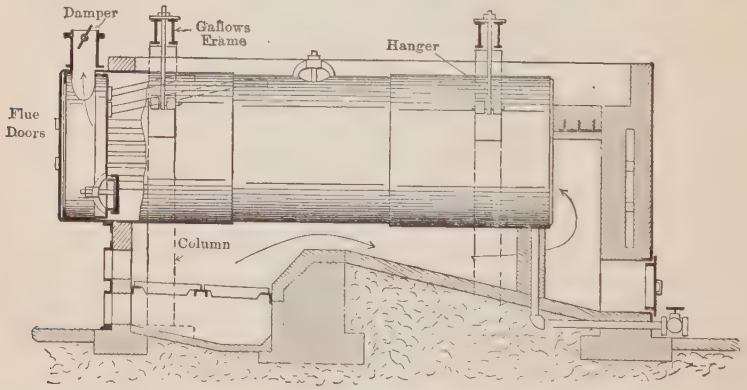


Fig. 358. — Horizontal Return Tubular Boiler with "Half Front."

avoid air leakages, which have a detrimental effect on the draft and boiler performance.

(b) An externally fired boiler classified as of *Horizontal Return Tubular type* ("H.R.T. boiler") is shown in Fig. 358. In this

boiler the smoke box is formed in an extension of the boiler shell which projects beyond the brick front wall. The cast iron "boiler front" covers only the portion of this wall located below the smoke box, and is therefore commonly called a "*half-front.*" The boiler shown in this figure is suspended from cross beams or "Gallows frames."

(c) In Fig. 359 is shown a H.R.T. boiler with "*full flush front,*" the smoke chamber being formed in the brickwork of the front wall. The boiler is shown to be supported by brackets,

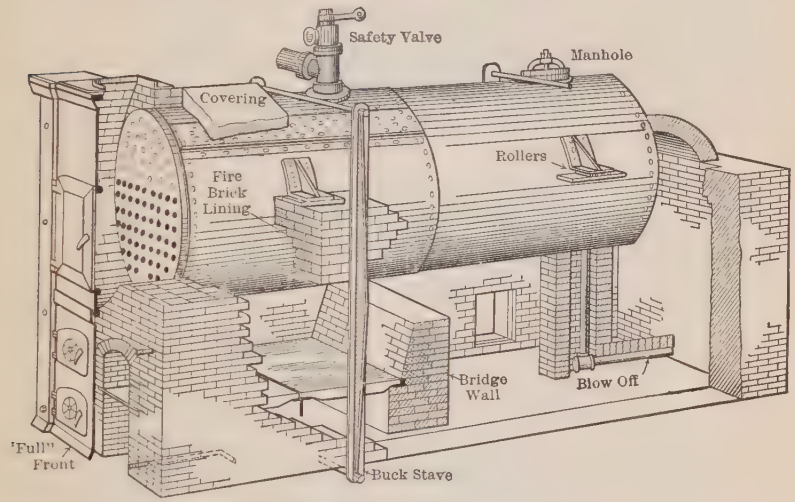


Fig. 359.—Horizontal Return Tubular Boiler with "Full Flush Front."

the rear pair of which is mounted on rollers to allow free expansion. The brick setting is braced by "buck staves."

As most of the scale is deposited at the rear of the boiler, as in Fig. 350 (*a*), the blow off is located at this point. The back end of the boiler is lowered slightly to aid in draining the shell.

As the boiler shell is exposed to the direct heat of the furnace, the thickness of metal must not be so great as to make it liable to burn thinner, and as the shell thickness must increase with the diameter of the boiler, these boilers cannot be constructed beyond a certain size. They are not ordinarily built larger than 200 boiler horse power and are seldom used with pressures above 150 pounds. The H.R.T. boilers are about the cheapest made, hence are quite widely used for low pressures.

267. Water Tube Boilers. (a) Figs. 360 and 361 illustrate sectional water tube boilers known as the *Babcock and Wilcox type* (or "B. and W." type). The tubes are expanded into pressed steel front and rear "headers" to form tube "sections." The sections, the drums, connecting nipples and other parts are shipped "knocked down," and are assembled at the power house. The parts to be transported and handled are therefore relatively small. Being of the sectional type with small elements, the danger of disastrous explosions is slight, as ruptures seldom occur

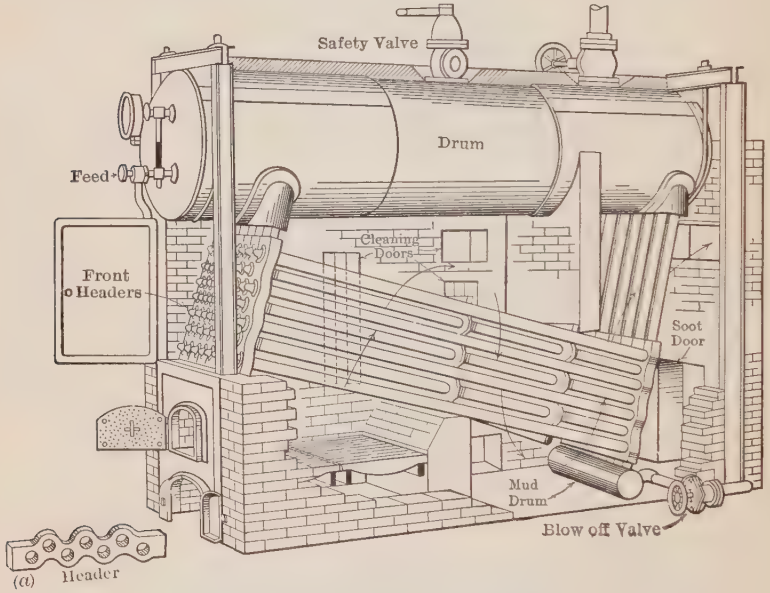


Fig. 360. — B. and W. Type of Boiler.

elsewhere than in the tubes. Opposite each tube is a hand-hole cover which can be removed for cleaning and replacing tubes. Doors for external cleaning are provided in the side walls, hence these boilers cannot be arranged in continuous batteries, but they may be grouped in batteries of two each. The boiler has elements similar to (*b*) in Fig. 350, with mud drum located at the bottom of the rear headers. It is hung from above, hence is free to expand or contract.

As shown in the figure the furnace has an exposed roof and the gases are baffled so as to make three "passes" across the tubes.

Other arrangements of baffles and furnace can of course be used. The tubes are not arranged in vertical rows, but are " staggered,"

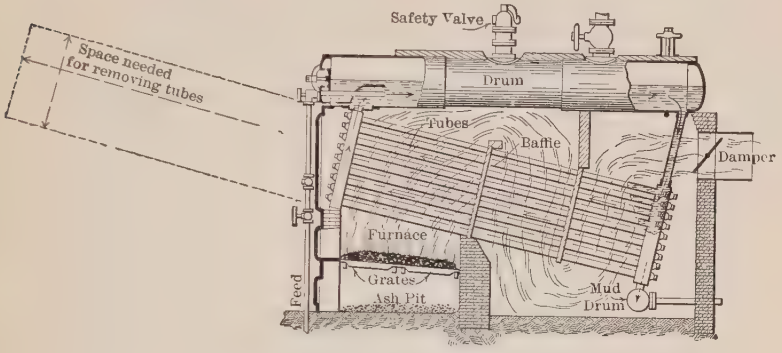


Fig. 361. — B. and W. Type of Boiler.

as shown by the header at (*a*) in Fig. 360, so as to further baffle the gases.

(b) In Fig. 362 is shown the *Heine type* of water tube boiler having the front and rear " water legs " made of steel plates and

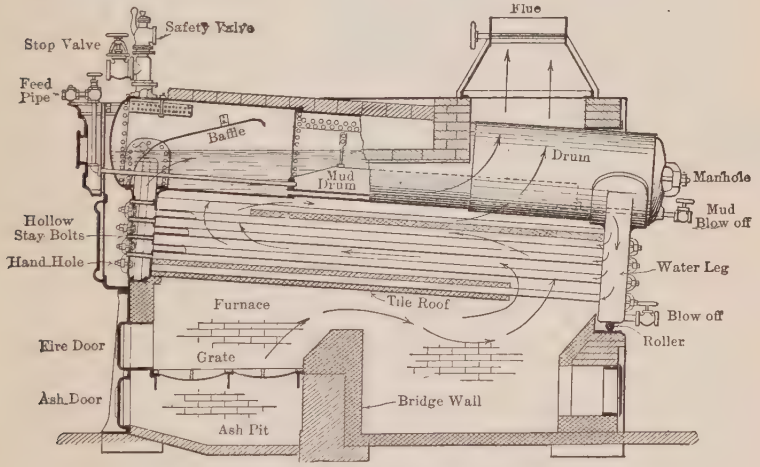


Fig. 362. — Heine Type of Boiler.

riveted to the drum. The front and back plates of each water leg are held together by hollow stay bolts having holes large enough to permit the insertion of a steam or air pipe for blowing the soot and dust from the exterior of the tubes; and opposite

each tube in each water leg is a hand hole giving access to the interior of tubes.

The feed water enters a "mud drum," where it remains quiescent for a considerable time before it mixes with the water which is circulating through the tubes. As this feed water becomes heated certain of the impurities are precipitated in the mud drum, from which they can be blown off at intervals.

The water legs of the boiler shown in the figure rest on the brickwork and have rollers under the rear one. The boiler may, of course, be supported in other ways. This boiler is shipped completely assembled, ready to have the setting constructed immediately upon its arrival. As no cleaning doors are located in the side walls, such boilers can be arranged in continuous batteries.

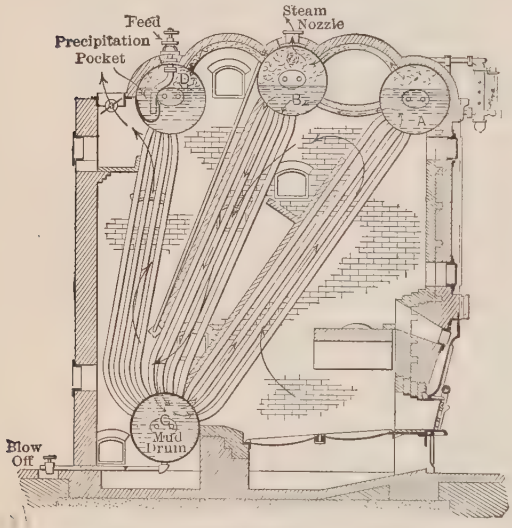


Fig. 363. — Stirling Type of Boiler.

As shown, the furnace has a tile roof supported by the lower row of tubes and the furnace gases are baffled so as to pass *along* the tubes. The same type of boiler is often used with baffles arranged similar to those in Fig. 361.

(c) Fig. 363 shows the *Stirling type* of water tube boiler which may be classed as a vertical one. It is composed of drums and tubes which are not assembled until they reach their destination. Since the elements are simple and easy to make, the cost of such boilers is less than that of those having more complicated parts. The feed water enters a precipitation pocket in drum *D* and is heated as it descends to drum *C*. The circulation of water is through tubes joining drums *A*, *B* and *C*. All the upper drums are connected by steam pipes.

The rows of tubes running circumferentially around the drums

are arranged in pairs, between which is sufficient space for the removal or insertion of tubes located in the interior of the nests. The tubes are curved and the mud drum is suspended by the tubes — an arrangement which gives flexibility and permits of expansion and contraction accompanying temperature changes. All drums have man holes which give access to their interiors and to the tubes. The side walls of the setting are provided with cleaning doors exposing the exteriors of the tubes, hence

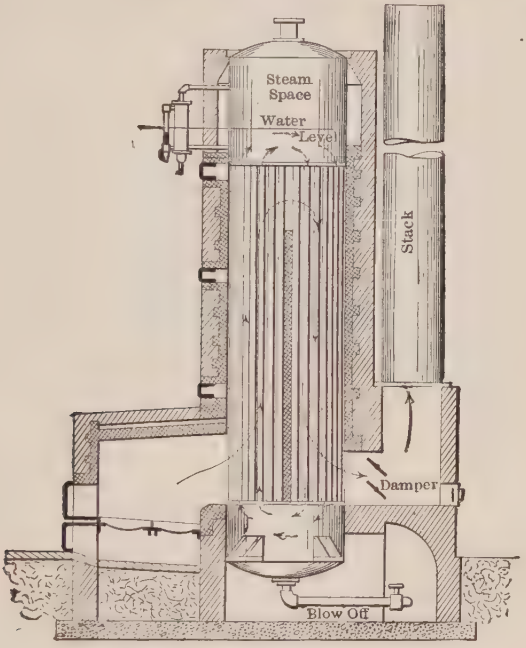


Fig. 364. — Wickes Type of Boiler.

these boilers cannot be arranged in continuous batteries — they may, however, be arranged in pairs. The arrangement provides for large combustion space and for an ample coking arch, or Dutch oven roof.

(d) There are many other arrangements of boilers composed of simple horizontal drums and vertical tubes. In some the vertical tubes enter a single upper drum and a single lower one, and the gases are baffled so as to make two or three passes.

In other boilers there are two drums above, with steam and water connections between, and two lower drums, with water

connections, and between these pairs are vertical tubes. The furnace gases pass up along the tubes joining the front drums and down along the rear tubes, to a flue connection near the bottom of the setting. There are still other arrangements, but those given will suffice to show the possibilities of this construction.

(e) Fig. 364 shows the *Wickes type* of vertical water tube boiler having single upper and lower drums with vertical axes. The tubes are removed and inserted through hand holes located in the dome of the steam space; hence, it is necessary to provide overhead room, or sky-lights, immediately over the boilers.

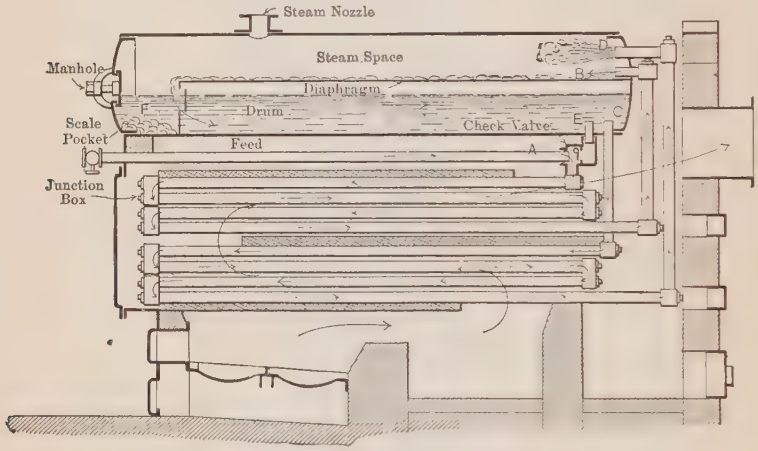


Fig. 365. — Parker Type of Boiler.

(f) In Fig. 365 is shown the *Parker type* of boiler, which differs radically in several respects from the ordinary types, since it makes use of the counter-flow principle (Sect. 261 (*i*)) and delivers the steam and water at points above the water level (Sect. 261 (*l*)). Under the best conditions of operation, the feed water enters the system at *A* (check valve *E* being closed), passes downward through the zig-zag tubes in a general direction opposite to that of the ascending flue gases (just as in Fig. 349) and is finally delivered at *B* into the drum, with a temperature high enough to cause the precipitation of most of the impurities. The water from the drum enters another set of zig-zag pipes at *C*, constituting the vaporizing element, is further heated as it descends and is finally discharged as steam into the drum at *D*.

Owing to the countercurrent flow it is possible, by providing sufficient heating surface, to cool the flue gases below the temperature of the steam. Hence, sometimes a third nest of tubes is added, in which case the upper is called an "economizer element."

Should the feed valve be shut off the circulation will still continue, for water will then pass through check valve *E* into the upper element of tubes, but the operation will be somewhat less efficient than before.

The "junction boxes" joining the ends of the tubes have hand holes and some have non-return valves which ensure the circulation of the water in the proper direction. Besides the scale pocket *F* there is a blow off (not shown) located under the diaphragm in the bottom of the drum.

(g) In boilers of the *porcupine* type, the tubes, with closed outer ends, project from a water drum, or header, into the path of the flue gases. Fig. 366 shows an arrangement having a double header and concentric tubes. With such arrangement the cold water descends through the left water leg, passes through the inner tube to the end, then returns through the annular passage between the two tubes and ascends in the riser to the steam drum above. But little use is made of this arrangement, however, because of the expense of construction.

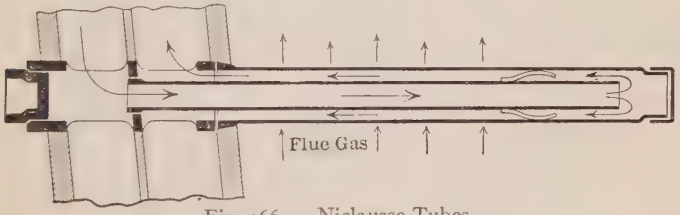


Fig. 366. — Niclausse Tubes.

(h) Fig. 367 shows a *double-furnace boiler* arranged to be fired from both front and "rear"; thus it has about double the grate area and generates steam nearly twice as rapidly as in the ordinary case. However, as the rate of evaporation is high the efficiency is slightly less than is obtained under less intensive conditions. Such arrangements are frequently adopted when floor space is limited or when the cost of real estate is great, as is the case in power plants located in congested districts of large cities. The same scheme can, of course, be used with boilers of other types than that shown.

268. Boiler Accessories. In addition to the fittings already described, boilers are always fitted with steam gauges, glass water gauges, try cocks, safety valves, feed valves, blow-off valves and steam stop valves. In addition they frequently have "water columns" with floats to operate sentinel whistles when

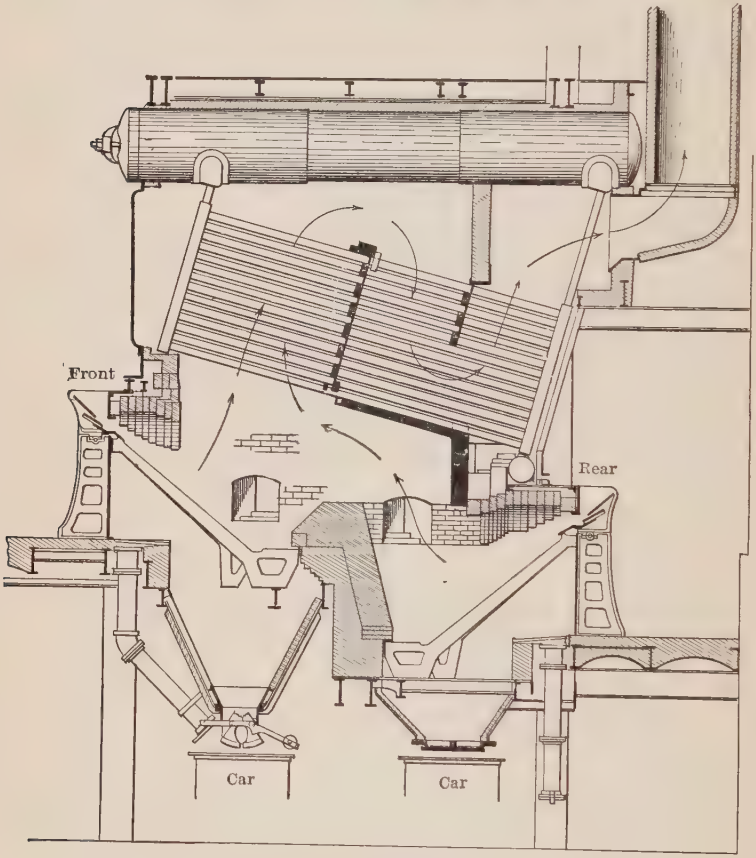


Fig. 367. — Double-furnace Boiler.

the water level becomes too high or too low; and they may have "fusible plugs" which, when the water level becomes dangerously low, become uncovered and melt, and thus allow steam to escape and attract attention before the plates become overheated. Automatic feed-water regulators are sometimes used, and in plants where the load fluctuates rapidly damper regula-

tors, controlled by the steam pressure, are also provided to automatically adjust the draft. When water tube boilers are used, the boiler room equipment generally includes mechanical tube cleaners, of which there are many varieties.

269. Boiler Performance. (**a**) In stating the evaporative performance of boilers and similar apparatus, it is customary to use the latent heat of vaporization of one pound of steam at atmospheric pressure as the "*Unit of Evaporation*" (U.E.). The value of this unit is, therefore, 970.4 B.t.u. (old value 965.7); and, as this is the heat absorbed by one pound of steam in being converted *from* water with temperature of 212° F., to steam *at* the same temperature, the performance may also be stated in terms of the "*Equivalent Evaporation*" or "number of pounds of water *from and at 212° F.*" that would be evaporated by the same amount of heat. Evidently there would always be the same number of pounds of Equivalent Evaporation as there are Units of Evaporation.

When boilers are tested, the temperature of the feed water, the actual weight of steam generated per hour, and the quality (or superheat) are all determined. With these quantities known, the equivalent evaporation from and at 212° F. can be determined by dividing the heat given to water and steam by 970.4 (965.7).

(**b**) This same unit can also be used for expressing the value of fuels when used for generating steam; thus, the *Theoretical Equivalent Evaporation* (T.U.E.) *per pound of fuel* is found by dividing the calorific value per pound by 970.4.

Based on the usual calorific values, the T.U.E.'s per pound of *combustible*, are about as follows for the different kinds of fuel: — Carbon, 15 pounds; good anthracite, 15.4 pounds; semi-bituminous, 16.3 pounds; bituminous, 14 to 15.8 pounds; and oils, 18.5 to 22 pounds.

Obviously the Theoretical Equivalent Evaporation per pound of *fuel* can be obtained by multiplying the foregoing figures by the percentage of combustible present.

(**c**) The steam generating apparatus as a whole delivers with the steam only a portion of the calorific value of the fuel, the percentage depending on the overall efficiency of the apparatus. These efficiencies were given in Sect. 260 (*h*) and (*i*). With good coal the *Actual Equivalent Evaporation* should be at least

9 pounds per pound of *combustible*, and in the best instances $12\frac{1}{2}$ pounds have been reached. With oil this evaporation is from 14.4 to 16.9 per pound of fuel.

When the *rate* of evaporation is under consideration, the unit of time generally adopted is the hour—hence the terms "Equivalent Evaporation per hour" and "Units of Evaporation per hour" are in common use. If no unit of time is specified the *hour* is implied.

(d) In boiler computations it is sometimes convenient to make use of a quantity called the "Factor of Evaporation" (F.E.). This quantity is the ratio of the heat absorbed per pound of steam generated to 970.4 (or 965.7 as the case may be). Hence

$$\text{Factor of Evaporation} = \frac{(q + xr + \overline{C}_p D)^* - (t - 32)}{970 \text{ (or } 965.7)}. \quad . \quad (406)$$

in which q, x, r, D and $\overline{C}_p$ are respectively the sensible heat, quality, latent heat, degrees of superheat and specific heat of the steam leaving the boiler, and t is the temperature of the feed water. Evidently the Factor of Evaporation is the ratio of the equivalent evaporation to the actual weight, and as this ratio is frequently used its values are generally tabulated in reference books for different pressures of dry saturated steam with various temperatures of feed water.

(e) The rated size and the maximum capacity of boilers are usually stated in terms of a unit miscalled a "**Boiler Horse Power**" (B.P.). It has been suggested that this be changed to "*Boiler Power*," as the term "horse power" is inapplicable to boilers. The Boiler Horse Power is defined as the equivalent of 34.5 pounds of steam evaporated "from and at" 212° F. per hour; (i.e., $34\frac{1}{2}$ U.E. per hr.). It is, therefore, merely a measure of the heat given to the water and steam and is equivalent to the transfer of 33,479 B.t.u. per hour (with U.E. = 970.4 B.t.u.).

The "horse power" of a boiler which is evaporating a given weight of steam per hour at a certain pressure, with a certain quality, and from feed water at a certain temperature, can therefore be found in two ways: — First, by dividing the equivalent evaporation by $34\frac{1}{2}$; second, by dividing the total heat supplied

* The expression in this parenthesis is written so as to apply to both superheated and saturated steam, as was first done on page 173.

to the water and steam per hour by the number 33,479 given above.

(f) In this connection, however, it is important to note that there is no definite relation between engine horse power and the so-called boiler horse power; the ratio of the engine h.p. to the "boiler h.p." in any plant depends entirely upon the economic performance of the engine, hence it is not necessarily the same in different plants.

270. Proportioning the Boiler for Power Output. (a) The American Society of Mechanical Engineers has made the following recommendation regarding boiler ratings. "A boiler rated at any stated capacity should develop that capacity when using the best coal ordinarily sold in the market where the boiler is located, when fired by an ordinary fireman, without forcing the fires, while exhibiting good economy. And further, the boiler should develop at least one-third more than the stated capacity when using the same fuel and operated by the same fireman, the full draft being employed and the fires being crowded, the available draft at damper, unless otherwise understood, being not less than one-half inch water column."* Boilers of the commercial types generally have overload capacity considerably in excess of the $33\frac{1}{3}$ per cent here specified. Some boilers are being operated continuously under loads double those for which they were originally intended, and triple outputs have been obtained in a few instances.

(b) The total amount of heating surface needed by boilers can be determined either by multiplying the boiler "horse power" by the number of square feet needed for developing one horse power, or by dividing the total equivalent evaporation per hour by the allowable rate of evaporation per square foot.

(c) Most stationary boilers of the "water tube" type have 10 square feet of heating surface per boiler horse power under normal load, the corresponding rate of equivalent evaporation per square foot per hour being about $3\frac{1}{2}$ ($= 34.5 \div 10$); while the more common types of stationary "fire tube" boilers usually have 12 or more square feet per boiler horse power, the equivalent evaporation being 3 pounds per square foot or less. However values both larger and smaller than these are sometimes used.

* Trans. A. S. M. E. 1899.

When there are limitations as to space or weight, less heating surface and higher rates of evaporation are used. For example, in marine boilers from 4 to 8 square feet of heating surface are provided, the corresponding evaporation being from 8 to 4 pounds per square foot per hour, and in some instances marine boilers of the water tube type have been operated continuously with average rates as high as 16 pounds.*

(d) The fuel needed per boiler horse power hour can be readily determined by dividing $34\frac{1}{2}$ by the equivalent evaporation per pound — or by dividing 33,476 by the actual calorific value per pound corrected for boiler and grate efficiencies. Thus the *combustible* required per boiler horse power ordinarily ranges from 3 to 4 pounds per hour, depending on the kind of coal, and the weight of *coal* is roughly from 3.5 to 5 pounds.

* Melville, Engineering Magazine, January, 1912.

CHAPTER XXXI.

SUPERHEATERS.

271. Advantages of Superheating. (a) It has already been shown (Sect. 128 and 180j) that the water rates of steam engines and turbines may be materially improved by the use of superheat, but that the improvement in steam consumption is not a correct measure of the gain effected, since one pound of superheated steam contains more heat than an equal weight of saturated steam at the same pressure. Leaving out of consideration, for the time being, certain incidental advantages of superheating, the *true* measure of gain is on the basis of the *heat economy* resulting from its use and this is given by the ratio of the number of heat units supplied to the superheated steam, per horse power delivered by the engine or other prime mover, to that used when saturated steam is the working substance.

(b) In addition to such gains as may be effected in the prime movers themselves by the use of superheat, there may be a two-fold reduction in the heat lost in the connecting pipe lines, because (1) superheated steam loses heat much less rapidly than does wet steam and because (2) the radiating surfaces of the pipes may be made less — for smaller pipes can be used, as superheated steam may be allowed to flow at higher velocities than are permissible with saturated vapor.

(c) But the *ultimate* test of the advisability of installing additional apparatus, such as superheaters, is always on the basis of the *financial economy* effected. In the case in question the addition of the superheaters may not increase the total expense for the power plant equipment, for the improvement in heat economy may permit a reduction in the size and cost of the boilers, and the diminution of the water rates may make possible a decrease in the size and cost of the condensers and other auxiliary apparatus. Then, the operating expenses may be reduced not only by the saving in the expenditure for fuel but also by the reduction in the outlay for purchasing and pumping

the water used for feed and for condensation. To offset the gains is the additional expense involved in the operation and maintenance of the superheaters. The use of the smaller pipe lines, which are permissible with superheated steam, may not effect a saving in their cost, as the materials, construction and fittings must be of better quality than is required when saturated steam is used.

272. Types of Superheaters. (a) There are two general types of superheaters — (1) separately fired superheaters, and (2) built in, or boiler draft superheaters.

The first class is installed in a separate setting of its own and receives hot gases from its own furnace. The second class is located inside of the boiler setting and in line with one of the " passes " of the products of combustion.

(b) In each type the saturated steam, generally containing from 2 to 4 per cent of moisture, is led from the steam nozzle on the drum of the boiler, through the superheating apparatus on its way to the steam consumer.

(c) Superheaters of both types generally consist of a multiplicity of elements containing a small volume but exposing a relatively great surface. There are, however, several superheaters in which a few very large elements are so constructed that, by means of baffles or equivalents, the steam flowing through them is divided up into thin streams in contact with extended wall areas.

(d) Generally the metal used is mild steel, and the elements are composed of seamless tubes which are of small diameter (1 inch to $1\frac{1}{2}$ inch bore) with thick walls (0.15 to 0.2 inch thick) and which are connected with built-up, forged, or cast steel headers or their equivalents. In a few instances cast-iron elements with comparatively thick walls are still used, but there is a growing tendency to look with suspicion on the use of this material in cases where temperatures and pressures are high and where temperature changes are great.

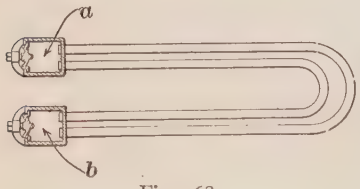


Fig. 368.

Figs. 368 and 369 show the two elements most commonly used in this country. Instead of having the tube ends enter separate headers, they are sometimes

connected with a single one arranged with suitable partition plates or baffles. The element shown in Fig. 369 has a thin annular steam passage between a sealed inner tube and an outer one which is surrounded by flanges. The flanges, which are of cast iron, present large heat-absorbing surfaces to the hot gas, protect the steel tubes and store heat, but add to the expense of construction. The steam is brought intimately into contact with the walls of the larger tube, since it can flow through the thin annular passage only.

(e) Experience has shown that the ideals to be attained in superheater construction and arrangement are: (1) perfect freedom of expansion; (2) ability to withstand high temperature, high pressure, and violent changes in temperature; (3) avoid-

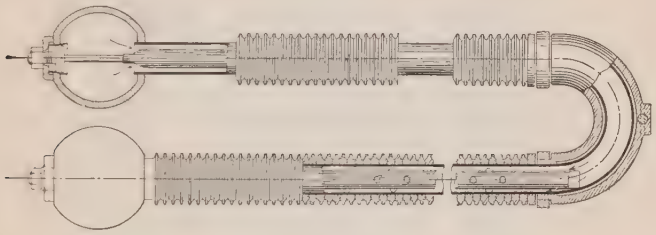


Fig. 369.

ance of screwed joints; (4) the protection of all joints from exposure to the hot gases; (5) provision for cleaning externally and internally; (6) means for adjusting the superheat to any desired temperature; (7) natural, or automatic, regulation to maintain that temperature; (8) means of bypassing the steam around the superheater when the latter is out of commission; (9) provision for flooding the elements (in some cases) with water and for draining them; (10) small space requirements; (11) low first cost; and (12) small expense of operation and maintenance.

273. Separately Fired Superheaters. (a) Two examples of separately fired superheaters are illustrated in Figs. 370 and 371. In all such apparatus it is nearly always necessary to prevent the flame and very hot gases from impinging directly on the superheating surface, it being generally considered that temperatures of from 1300 to 1500° are the highest allowable for the gases which are in contact with such surfaces; hence, the use of intercepting brick arches and walls through which the hot gases must

pass, as shown in Fig. 370, though a greater degree of security is attained by combining a water element with these walls, as shown in Fig. 371.

(b) The temperature of superheat may be controlled directly by varying the rate of combustion, by means of a damper, as in Fig. 371; by bypassing the gases, or by both of these methods, as in Fig. 370. But even if the dampers be made to normally follow the delivery temperature exactly, as can be done by means of thermostatic control, the heat stored in the walls of the setting will cause an abno mal rise of temperature when the

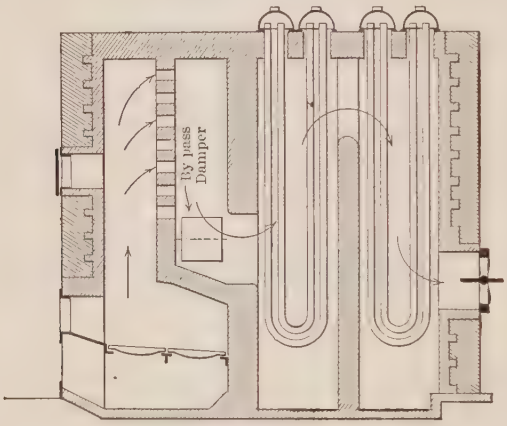


Fig. 370.

demand for steam suddenly decreases to any considerable extent. Then there may also be sudden drops in the temperature due to the inflow of cold air when the furnace doors are opened for firing.

(c) Compared with the built-in type, the separately fired superheater has many disadvantages, of which the principal ones are: (1) Greater first cost because of the separate setting and grate; (2) larger maintenance cost because of separate setting; (3) greater cost of operation because of separate furnace to be fired; (4) greater floor space occupied; (5) grate losses, which in this case are added to those of the boiler; (6) lower efficiency because the flue gas enters the stack at a temperature which must be higher than with built-in type, where superheater is followed by water heating surface; (7) greater radiation loss

because of individual setting; and (8) difficulty of controlling temperature of steam, as explained in (b) above. The separately fired superheater has the advantage that the boilers can still be used to supply saturated steam even when the superheater is out of commission; that it permits the variation in the degree of superheat to be made independently of the operation of the boiler, and one superheater can be used for several boilers.

(d) Although it has many disadvantages, the separately fired apparatus may be of value in many instances. In some plants, steel mills for instance, there are often large quantities of hot gases which, by such apparatus, can be used to superheat the steam coming from the boilers, but which would otherwise

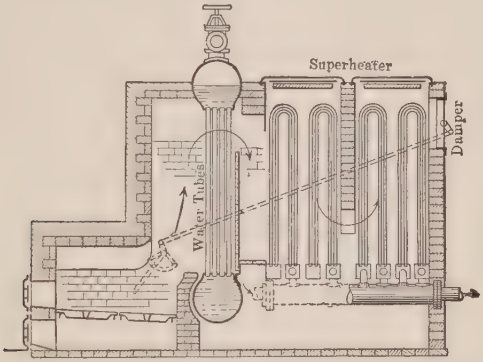


Fig. 371.

be wasted. Then, there are also many industries in which steam exhausted from engines is used in some manufacturing process, and in many such cases it is desirable to superheat this steam in separately fired superheaters. Again, either as a means of improving the economy or of increasing the capacity of a boiler plant already installed, it may be desirable to superheat the steam generated, and in such cases it will generally appear upon investigation that the separately fired unit is the better investment, as it will involve least changes in piping and settings.

274. Boiler Draft Superheaters. (a) Examples of this type are illustrated in Figs. 372 to 374. In nearly all cases built-in superheaters are installed at such a point in the flues, or gas passes, that the temperature of the gas reaching them can never greatly exceed about 1500° F. There are a few instances,

however, as in Fig. 372, in which the superheaters are installed in a separate brick chamber within the boiler setting and are supplied with hot gases directly from the furnace, by means of a passage in the brick walls of the boiler setting, the flow of gas being controlled by a damper in the passage.

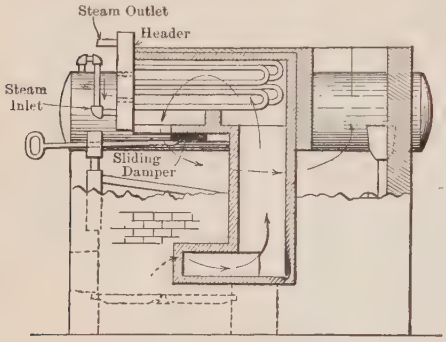


Fig. 372.

(b) Two distinctly different methods of maintaining an approximately constant temperature of superheat are in use. In one, the superheating elements are located at such a point (as in Fig. 373) that the gases reaching them vary in temperature and quantity as nearly as possible in proportion to the amount of steam flowing. The attainment of such conditions is generally more ideal than real, but is fairly well approximated in a few instances, since the amount of steam gen-

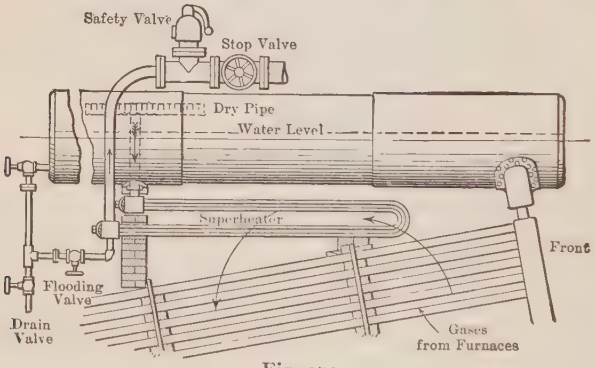


Fig. 373.

erated depends directly on the quantity and temperature of the gases coming from the furnace.

In the other method the superheating elements are installed within a separate chamber, as in Figs. 372 and 374, and a damper, which regulates the supply of hot gases, is put under some sort of control, which may be thermostatic. These superheaters have a certain temperature lag, as do the separately fired variety,

but it is not as great as in that case because of the smaller amount of brickwork surrounding them.

275. Protection of Superheater. (a) No superheater, no matter what its construction, will last for any considerable length of time if exposed to the hot furnace gases when steam is not flowing through it. To prevent damage in this way, during the period of firing up and when cooling down or standing idle, some protective device is essential.

(b) With separately fired superheaters the hot gases may be deflected, as in Figs. 370, 372, and 374, so that they bypass the superheater and flow directly from the furnace to the stack, or protecting rings like those in Fig. 369 may be used, or provision may be made for "flooding" the superheater — that is, filling it with water whenever the flow of steam ceases. The latter method is open to the objection that scale-forming material may be deposited in the superheater, thus decreasing its ability to transfer heat from gases to steam, which would ultimately result in maintaining the metal of the superheater at too high a temperature when in operation and thereby shortening its life.

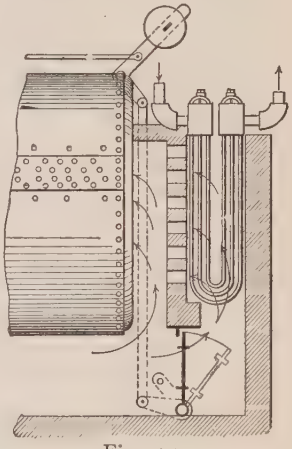


Fig. 374.

(c) When boiler draft superheaters are located in a separate chamber within the boiler setting, either of the above methods may be used, but the objection to the last holds equally well for this case.

(d) When boiler draft superheaters are located directly in one of the passes the most customary method of protecting is by flooding. They are generally so arranged (as in Fig. 373) that when flooded they form part of the boiler evaporating or heating surface, practically being connected in parallel with it, but so that they can be drained and connected in series with the boiler when superheat is desired.

(e) In connection with this latter arrangement, an auxiliary safety valve is sometimes placed between the superheater and the main stop valve, so that if this latter valve is suddenly closed,

or if the demand for steam suddenly ceases, before the fires can be deadened, the rising pressure of steam will pop this safety valve (before the main safety valve opens), and allow steam to pass through the superheater, thus protecting it temporarily and warning the attendant of the necessity for checking the fire and, possibly, for flooding the apparatus.

276. Superheater Surface. (a) The determination of the amount of surface required by a superheater to give a definite degree of superheat, when fired in a certain way, or located in a given position, is largely a matter of experience with each manufacturer of each different type of boiler. There are several distinct methods of approximating the amount of super-heating surface required; the three most common are given below.

(b) The superheating surface may be taken as a *multiple of the grate surface.* Thus for water tube boilers, the heating surface of built-in superheaters is generally taken at from 8 to 12 times the grate area, depending upon location within the setting, average rate of firing, superheat desired, character of coal, etc. With long flaming coals the gases often arrive at the superheater at a higher temperature than with short flaming fuels and a smaller surface may therefore be used.

For internally fired boilers, values between 25 and 35 times the grate area are used.

(c) The superheating surface may be taken as a *fraction of the boiler heating surface.* For water tube boilers it varies between 10 and 40 per cent of the boiler heating surface, though it rarely exceeds 20 to 30 per cent. For internally fired boilers a greater ratio is required, reaching in some cases to values almost equal to the boiler heating surface, and seldom dropping below 50 per cent of that surface.

(d) The number of square feet of superheating surface (S) required may be determined by calculations, taking account of *rate of heat transmission* per square foot per hour, which depends both on the coefficient (K) of heat transmission and mean temperature difference (θ_m) between steam and gases. In this case, the heating surface required to transmit per hour an amount of heat equal to ΔQ is

$$S = \frac{\Delta Q}{K\theta_m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (407)$$

The coefficient (K) is the heat (B.t.u.) transmitted per square foot of surface, per degree difference of temperature, per hour. Its value varies widely with conditions and is found by experiment or experience. High velocity, thin streams of steam or gas, violent agitation, and high temperature and pressure, increase its value. The condition of the superheater also has considerable effect; when scaled internally and covered with ash and soot externally, the rate of transmission is very low.

In general, values of K vary from 1 to as high as 10, and considerable experience is required in choosing a proper value.

CHAPTER XXXII.

DRAFT AND DRAFT APPARATUS.

277. General Principles. (a) The flow of air and products of combustion through the steam-generating apparatus is retarded by the resistances encountered in the various portions of the passage. The total resistance (R), from the point where the air enters the boiler setting to the base of the stack, is the summation of the resistances of the fuel bed (R_f), of the boiler passages (R_p), of the flues or breeching (R_b) and of any other passages (R_x) (such as that through an "economizer") which are traversed. Thus $R = R_f + R_p + R_b + R_x$. It is of course desirable to have the total resistance as small as possible, hence each component resistance should be reduced to the fullest extent allowable.

(b) As the gases flow only from places of higher pressure to those of lower (through a process of expansion) the gas pressures must decrease progressively from the point of entrance to the point of exit, the pressure drops through the different portions depending on the respective resistances of the parts.

The way in which the pressure varies, as the particles of gas advance through the steam-generating apparatus, is shown in a general way in Fig. 375 by the curve *abcd*, in which the abscissas represent the decrease of pressure below that at the point of admission and the ordinates are distances along the passage, measured from the same point.

Evidently, the same curve would apply (in this particular case) regardless of whether the inlet pressure is atmospheric, or greater or less than that, for the same pressure drops and the same gradients to the curve would still be required. The difference between the abscissas of any two points on this curve gives the pressure drop required to overcome the resistance between the corresponding points in the apparatus. The final abscissa represents the pressure drop developed through the whole apparatus, and is evidently equal to the summation of all the pressure drops, i.e., $\Delta P = \Delta P_f + \Delta P_p + \Delta P_b + \Delta P_x$, in which

the subscripts refer to the same parts as before. It is the function of the stack, or other draft-producing device, to develop this difference in pressure.

(c) As the pressure variations of the flue gas, measured from atmospheric pressure, are very low, they are ordinarily determined by means of water manometers, as shown in Fig. 375, and are therefore commonly expressed in terms of inches of water column ("hydraulic inches") as compared with atmospheric pressure.*

The total pressure drop from air inlet to base of stack is generally between 0.4 and 1.2 inches of water when determined in this way.

The velocity of flow is generally so small that the velocity head can be neglected in the ordinary problems that arise in connection with the subject under discussion.

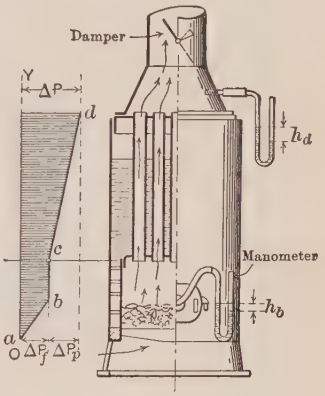


Fig. 375.

(d) According to *Bernoulli's theorem* (which can be applied to the steady and continuous flow of gases in long pipes when there are small pressure drops), the total head is the same at all points in the passage. At the point (o) of entrance it is the sum of the velocity head $\left(\frac{V^2}{2g}\right)_0$, the pressure head $\left(\frac{P}{\delta}\right)_0$, and the potential head z_0; and at any subsequent point (x) it is the sum of the similar heads for that point together with friction head F of the intervening passage. Thus

$$\left(\frac{v^2}{2g}+\frac{P}{\delta}+z\right)_0=\left(\frac{v^2}{2g}+\frac{P}{\delta}+z\right)_x+F, \quad . \quad . \quad (408)$$

in which v = velocity of flow in feet per second,
P = pressure in pounds per square foot,
δ = specific density = weight per cubic foot of gas, and
z = elevation in feet.

The friction head (ft.) is

$$F=f\frac{v^2}{2g}L\times\frac{S}{A}, \quad . \quad . \quad . \quad . \quad . \quad (409)$$

* One inch of water column corresponds to $\frac{62.5}{12}=5.2$ pounds per square foot; or one pound per square foot corresponds to .192 inches of water column.

in which L = length of flue in feet,
f = coefficient of friction,
S = length of perimeter of the cross-section in feet,
A = area of passage in square feet,

and the ratio A/S is called the "*mean hydraulic radius.*"

In the passages through the boiler the variation in velocity and in elevation can ordinarily be neglected, hence the quantities v and z disappear from Eq. (408) in this case.

Then from Eqs. (408) and (409), using a mean density δ_m and letting $\Delta P = (P_0 - P_z)$, the change in pressure head is

$$\frac{\Delta P}{\delta_m} = F = f\frac{v^2}{2g}L\frac{S}{A}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (410)$$

which shows that the pressure drop is dependent solely on the frictional resistance, which varies directly with v^2, L and the character of the surfaces and inversely with the mean hydraulic radius (A/S).

The velocity of flow is, from Eq. (410),

$$v = \sqrt{2g\Delta P \times (A \div fLS\delta_m)} \quad . \quad . \quad . \quad . \quad (411)$$

and for any given passage

$$v = \text{const}\sqrt{\Delta P}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (412)$$

Evidently the rate of combustion, which is dependent on the velocity (amount) of the air passing through the fuel bed, can be reduced by decreasing A in Eq. (411), other things remaining the same, as by partly closing the damper.

(e) But in the *actual case* of flow of gases through steam-generating apparatus, the conditions are quite different from the hypothetical ones assumed in connection with Bernoulli's Theorem, — for, through part of the passage there is air of a certain density, through the rest is a complicated mixture of gases varying as to composition, density and temperature; the passages are circuitous, have sudden changes in areas and in direction and have eddy pockets; the resistance through the fuel bed is constantly varying and the flow of gas is neither steady nor necessarily continuous, — hence, the analysis of the laws governing the actual case is difficult and as yet these laws are not well established.

There are, however, a few general statements which can be made and which are more or less applicable to most cases; they

may serve as rough guides in approximating the solution of problems connected with boiler draft.* These are given in the following paragraphs.

(f) Other conditions remaining the same (temperatures, resistances, etc.), the weight (w) of air entering the furnace in a unit of time is dependent on the velocity of flow, and appears to vary about as the square root of the pressure drop (ΔP) through the passages.† As the rate of combustion (R) is directly dependent on the air supply it varies approximately in like manner, i.e., $R = \text{const.}\sqrt{\Delta P}$, where the constant varies with the size and kind of coal, method of firing and other conditions. Thus, doubling the pressure drop increases the rate to about 1.4 ($=\sqrt{2}$) its former value; and to burn fuel twice as rapidly as before involves nearly quadrupling the draft pressure.

(g) It is also approximately true that if the resistances remain unchanged, the pressure drop through any portion of the passage will remain the same fraction of the total, regardless of the variation in the over-all drop, that is, the pressure gradients would vary proportionally. For example, for the case shown in Fig. 375, with change in draft, the curve would merely be replotted with all abscissas changed proportionally to the variation in the over-all drop, or the same curve could be used with suitable change in scale.

(h) The resistances encountered vary about as the square of the velocity (as in Eq. 409), although probably the exponent should be slightly less than 2, say 1.8. Hence, doubling the velocity, to obtain a twofold rate of combustion, necessitates about four times as intense a draft, and nearly four times as much *work* will be done in moving the gases.

(i) As the *power* is the product of resisting force by the velocity of motion, the amount required for removing the gases varies about as the cube of the velocity of flow, i.e., as the cube of the amount of air supplied, or as the cube of the rate of combustion. Thus, in order to double the rate of combustion, or boiler output, the draft-producing apparatus would have to do nearly eight times as much work. Therefore, while from the standpoint of space occupied by the boiler it may be desirable to

* Bull. 21, U. S. Bureau of Mines, "Significance of Drafts," contains discussions of experiments on draft.

† Thus it follows approximately Eq. (412).

force the rate of combustion as much as possible, the additional expense for power and apparatus for handling the gases with greater velocity places a limit beyond which it is financially unprofitable to go.

278. Amount of Pressure Drop Required. (a) The pressure drops (h_f inches of water) generally needed for overcoming the resistance through the fuel bed have already been given in Fig. 326 for the different rates of combustion of several sizes of various kinds of coal when burned under the usual conditions. But much variation from these curves exists, since the intensity of draft depends also on the thickness of the fuel bed, character of the ash and clinker, method of firing and other items.

(b) The drop in pressure (h_p inches of water) through the boiler passages depends on the length and cross sections of passages, arrangement of baffles, arrangement of tubes, etc. Under ordinary rates of combustion it ranges as in Table XXIV and varies about as the square of the rate of combustion (as explained in Sect. 277 (f)) when operating at greater or lower rates.

TABLE XXIV.—PRESSURE DROPS THROUGH BOILERS.*

B. and W.—double deck.....	0.4 in.	Stirling or Heine.....	0.2 in.
B. and W.—standard.....	0.3 "	Return Tubular.....	0.1 "

(c) To overcome the resistance of the breeching, or flues, between the boiler and the stack, a pressure drop (h_b) of about $\frac{1}{10}$ inch of water is generally assumed per 100 feet of length of smooth round passage when the boilers are being forced, and half as much for each elbow, though much depends on the mean hydraulic radius $\left(\frac{A}{S}\right)$ of the passage, on the curvature of the bends, character of wall surface, etc.

It would of course reduce the cost of the breeching if small cross sections and high velocity of flow were used. But since the resistance varies as the square of the velocity, greater draft would then be needed to overcome it, and this would, in general, add to the expense for the stack (or other draft-producing apparatus) an amount greater than the saving effected in outlay for the breeching. Hence, the breeching is usually given a liberal cross sectional area, one at least equal to that of the stack and gener-

* Kingsley, Eng. Record, Dec. 21, 1907.

ally 20 per cent greater, the velocity of flow being not more than that in the stack, and generally about 20 per cent less.

(d) The total pressure drop (h_1) inches of water between air inlet and base of stack evidently equals the sum of the drops through these various elements of the passage and of any others, such as those through economizers, which may be located between the boiler and stack. Thus $h_1 = h_f + h_p + h_b + h_x$. The draft-producing apparatus should of course be proportioned to give a pressure drop at least sufficient to cause the greatest rate of combustion that will ever be demanded with the poorest fuel which is likely to be used; then, smaller rates can be obtained by reducing the amount of air supplied, which can be done by regulating the dampers and air inlets either by hand or by some automatic device, the latter being generally operated by the slight variations in steam pressure which accompany the changes in the demand for steam.

(e) The current of gases through the boiler can be caused either by "*natural*" *draft* of a chimney (or stack), or by "*artificial*" *draft* maintained either by steam jets or by power-driven fans.

The duty of the draft-producing apparatus is twofold — first, it must produce the needed intensity of draft and, second, it must provide means for carrying off the products of combustion.

279. Chimney Draft. (a) When one pound of carbon is completely burned in air to CO_2, the latter gas will have the same volume at the same temperature and pressure as did the oxygen with which the carbon united (in accordance with Sect. 238 (a)); but the resulting flue gas will have one pound more material than was in the air which supplied the oxygen. Thus, for example, if the excess coefficient is two, 24 pounds of air with specific density = .0807 * are supplied for the complete combustion of one pound of carbon and there will result 25 pounds of gas, which will have a specific density * of $(\frac{25}{24}) \times .0807 = .084$. Hence the weight of a column of air one foot high, at sea level and at temperature t_a° F., is

$$D = (.0807 \times 492) \div (t_a + 460), \quad . \quad . \quad . \quad (413)$$

* Pounds per cubic foot at 32° F. (or 492° Absolute) and under 14.7 pounds square-inch pressure (measured at sea level).

and that of a similar column of flue gas, with excess coefficient equal to two and with temperature t_g° F., is

$$d = (.084 \times 492) \div (t_g + 460). \quad . \quad . \quad . \quad (414)$$

(b) In Fig. 376 (*a*) the intensity of pressure exerted on the side of the partition X by the column (A) of cold air at temperature t_a, and that exerted by the equal column (G) of hot gas at t_g° F., are respectively, in inches of water,*

$$h_a = .192\,HD = 7.64\,H/(t_a + 460), \quad . \quad . \quad . \quad (415)$$

and

$$h_g = .192\,Hd = 7.95\,H/(t_g + 460), \quad . \quad . \quad . \quad (416)$$

where H is the height of the columns in feet, and D and d have the values given in Eqs. (413) and (414). This is of course on

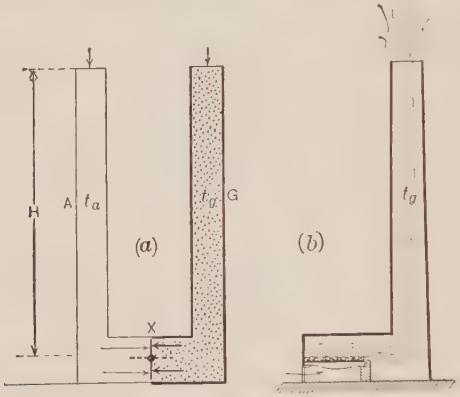


Fig. 376.

the assumption that equal pressures of air are exerted on the tops of these columns. The difference between these pressure intensities on the opposite sides of the partition is (at sea level)

$$h_t = (h_a - h_g) = H\left(\frac{7.64}{t_a + 460} - \frac{7.95}{t_g + 460}\right). \quad . \quad (417)$$

and, if the partition is removed, this will be the draft pressure tending to cause the flow of gases upward through column G.

(c) If means are provided for maintaining the high temperature (t_g) in column G, there will be a constant flow of gases, and as the air in column A is under atmospheric conditions the enveloping shell around that column can be omitted. Under these circumstances the conditions are those existing when a

* See footnote, page 575.

furnace and chimney (stack) are in operation, as in Fig. 376 (*b*); hence Eq. (417) can be used for obtaining the theoretical *draft pressure* h_t developed by a stack of height H feet when the resistances through the stack are neglected, and it gives the draft pressure that would occur when the ash-pit doors are closed and no gases are flowing.

Then, the *theoretical height* (H_t) of stack needed for producing a draft pressure of h inches of water at its base is (from Eq. 417)

$$H_t = h \div \left(\frac{7.64}{t_a + 460} - \frac{7.95}{t_g + 460} \right) \quad . \quad . \quad . \quad (418)$$

at sea level (14.7 lbs. per sq. in.). Obviously H_t will vary with changes in the atmospheric pressure (barometer).

(**d**) Under normal conditions the temperature of the flue gas at the base of the stack generally lies between 500 and 600° F. in different types of boilers; but if "economizers" are used, it will be less and in some instances may be reduced to 300° F. When the boilers are being forced the temperature rises above the normal, which helps to augment the draft. In using Eqs. (417) and (418) for the draft and the height of a new stack, t_g should be taken as the lowest flue temperature and t_a as the highest atmospheric temperature that are liable to exist simultaneously. As the gases become cooled in passing up the stack, t_g should be the mean temperature; it is customary, however, to use the temperature at the base of the stack and then to correct for the error, and for the resistances within the stack, by making the actual height about 25 per cent greater than the theoretical one. The effects of the column of hot gases above the stack and of the wind are generally neglected. Whether the wind assists or retards the draft depends on the arrangement of the chimney top.

In practice the height of stack is from 80 feet, with free burning coals and little resistance, to 175 feet, or more, with fine anthracite coal and with considerable resistance in the passages. But in settled districts the height should always be sufficient to satisfactorily dispose of the obnoxious gases.

(**e**) The *cross sectional area* of the stack should, of course, be made ample for accommodating the gases when the boilers are forced to their maximum capacity, and in fixing the size allowance should always be made for any possible growth.

Having found the actual height of stack, it is quite common practice to compute the cross sectional area by using *Wm. Kent's empirical formula*, which was derived as follows:—

Assuming that the volume of gas formed per hour is dependent on the amount of coal burned, which in turn is proportional to the boiler horse power (BP) developed, and that the velocity of flow varies as the square root of the height H (feet) of stack, it follows that the area is a function of $BP \div \sqrt{H}$. Then, from an analysis of numerous chimneys, Kent found that the *effective area* (E), in square feet, should be about

$$E = .3\,(BP) \div \sqrt{H}. \quad . \quad . \quad . \quad . \quad . \quad (419)$$

It is also assumed that if it is considered that the chimney has a two-inch lining of stagnant gas, the flow through the remainder of the cross section can be taken as being without resistance. Hence the *actual* diameter of a circular chimney and the length of side of a square one are made four inches greater than the corresponding dimensions determined for the effective area.

Kent's proportions are liberal as they provide for the combustion of about 5 pounds of coal per B.P.-hour, whereas not over 4 pounds are ordinarily used. They allow for velocities of gas through the stack ranging from about 20 ft./sec. with 100 feet of height to about 30 ft./sec. in a 200-foot stack.

(f) A more *rational method* of determining the proportions of a stack for a given set of conditions may be carried through in the following order:

1st. Assuming from 250 to 300 cubic feet of air at 60° F. as the amount needed to support the combustion of one pound of coal, and knowing the maximum weight of fuel to be consumed per unit of time, compute the corresponding total volume of gas at stack temperature.

2d. Assuming a velocity of flow of from 20 ft./sec., for short stacks, to 30 ft./sec., or more, for very tall ones, compute the effective cross sectional area needed to discharge this volume; and then, allowing for a two-inch lining of stagnant gas, determine the final dimensions of the cross section.

3d. Find the loss of draft (h_2 inches of water) arising from the stack resistances, which are due to (*a*) change of direction of the gases upon entering the base of the stack, (*b*) the skin friction and (*c*) the displacement of the atmosphere by the issuing

stream. From Kingsley's experiments * this loss for a velocity v ft./sec. was found to be given approximately by the equation

$$h_2 = .00036\, v^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (420)$$

4th. Determine the pressure drop h_1 up to the base of the stack and compute the theoretical height (H_t) from Eq. (418).

5th. Then find the actual height (H) of stack from

$$H = H_t\,(h_1 + h_2) \div h_1 \quad . \quad . \quad . \quad . \quad . \quad (421)$$

(g) By using the higher velocities, the stack diameter is decreased, which would result in a reduction in the cost of the stack if other things remained the same; but these greater velocities necessitate an increase in the height of stack, thus entailing an additional expense which either partly or wholly offsets that saving. Evidently for a given set of conditions there is some velocity which will give a proportion of height to diameter requiring a minimum amount of material for constructing the stack, and hence involving the least outlay of money.

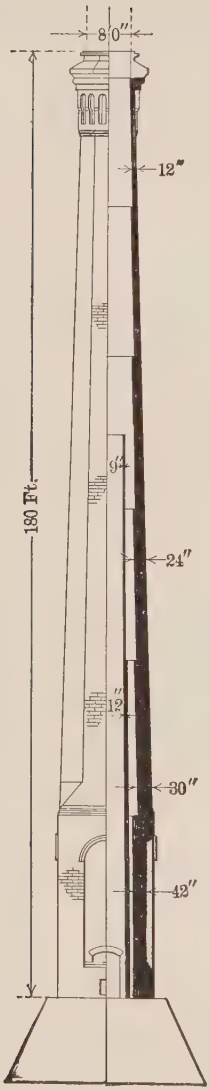
(h) For *rough estimating* it can be assumed that a 100-foot stack with gases at 500° and air temperature at 70° will exert a theoretical draft pressure of .6 inches of water at its base; that the draft varies directly with the height; and that the effective cross sectional area in square feet is equal to the number of pounds of coal burned per minute. For ordinary conditions with bituminous coal the stack area is about $\frac{1}{7}$th the grate area and with anthracite coal about $\frac{1}{5}$th.

(i) The different parts of a chimney and its foundation must not only carry the weights above but must also withstand the wind pressure. **Chimneys** are made of (1) common brick, (2) radial brick, (3) reinforced concrete and (4) steel plates.

A comparison of Figs. 377 to 380, which illustrate stacks of the different types but of the same height and internal diameter, will show roughly the relative thickness, weight, extent of foundation and space occupied with the various constructions.

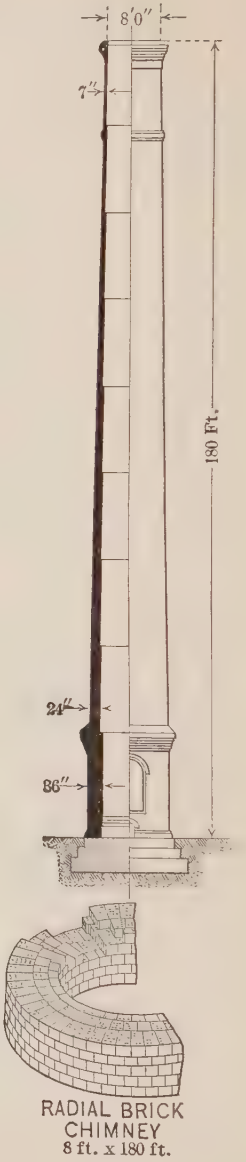
(j) If made of *ordinary brick* (Fig. 377) the chimney must be lined at least for part of its height with fire brick so set as to have perfect freedom to expand or contract with temperature

* Engineering Record, Dec. 21, 1907.



BRICK CHIMNEY
8 ft x 180 ft.

Fig. 377.



RADIAL BRICK
CHIMNEY
8 ft. x 180 ft.

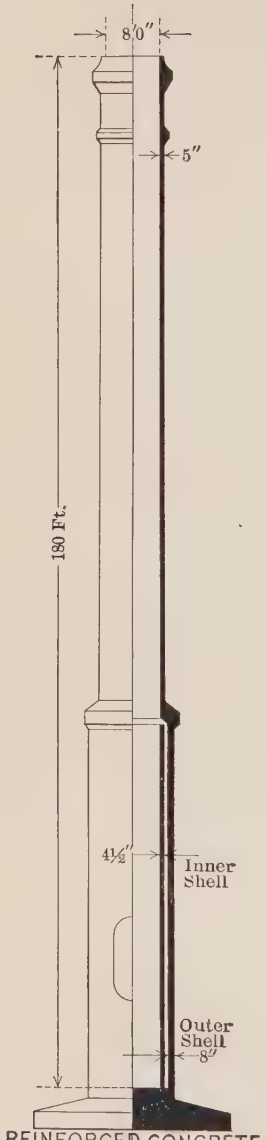
Fig. 378.

changes. By using special *radial brick* (Fig. 378), composed of suitable material, and commonly made perforated, (1) the lining may be omitted, (2) the shell may be thinner and of lighter weight, and (3) the foundation may be smaller; besides which (4) the construction is better and (5) can be more rapidly done than with ordinary brick. The tallest chimney in the world is of this type. It is located at Great Falls, Mont., and is 506 feet high with 50 feet diameter at the top.

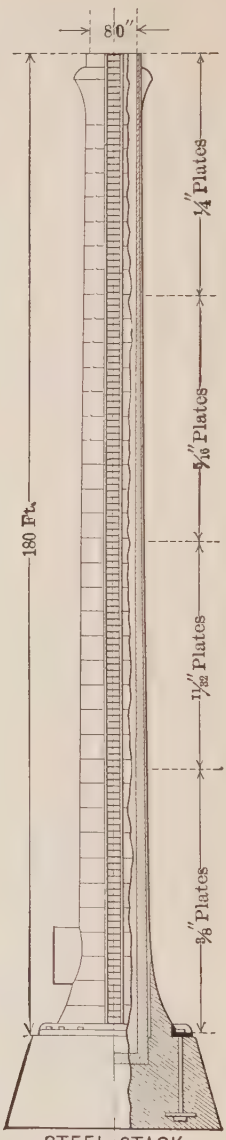
(**k**) Many chimneys are now made of *reinforced concrete* (Fig. 379), the steel reinforcing bars being arranged both circumferentially and vertically, the latter extending into the foundation, which is similarly strengthened. Such chimneys are (1) thinner than the brick, (2) weigh less, (3) occupy less space, (4) require but small foundations, (5) are free from joints and (6) can be rapidly constructed. The inner shell may be either of brick or reinforced concrete and in some cases is entirely omitted.

(**l**) In order to withstand the wind, *steel stacks* are either *guyed* with wire or wire rope, or else have flared bases bolted to the foundation, in which case they are said to be *self-supporting*. They are preferably lined with brick to protect the metal from the heat and corroding action of the gases. The lining may either be self-supporting or else be constructed in independent sections each resting on a bracket extending from the steel shell. Such chimneys are (1) of light weight, (2) easily and rapidly constructed, (3) cost little, (4) occupy small space (except when flared) and (5) are free from air leakage if properly calked. They must be painted frequently to protect the metal from the weather and from the gases.

280. **Artificial Draft.** (**a**) In a new power plant artificial draft apparatus is frequently employed as a substitute for a tall chimney, or to assist a short one, under the following conditions: (1) when the temperature of the stack gases is low, as when an economizer is used; (2) when the rates of combustion are high; (3) when fuels requiring intense draft are to be burned; (4) when certain stokers, like the underfed, are used; and (5) in certain other cases where in the long run it may be more desirable or more economical to purchase, operate and maintain such apparatus rather than have a chimney of large size.



REINFORCED-CONCRETE CHIMNEY
8 ft. x 180 ft.
Fig. 379.



STEEL STACK
8 ft. x 180 ft.
Fig. 380.

In an old plant it may be desirable to install artificial draft apparatus (1) to assist the original chimney when the plant has been increased beyond the capacity of the natural draft; (2) when it is desired to adopt unusual rates of combustion, or (3) to burn fuels requiring intense draft; (4) when there may be large emergency overloads or peak loads of short duration; and (5) when there are large and sudden changes in demand on the furnaces.

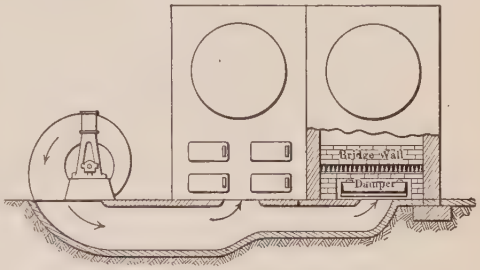


Fig. 381. — Forced Draft.

(**b**) In addition to its advantages in the instances already discussed, the artificial draft apparatus is (1) easily installed; (2) is transportable and (3) occupies but little space; and (4) it also permits of careful adjustment of the air supply, which makes possible more perfect conditions of combustion. The regulation of air can be automatic, the controlling device being operated by the slight changes in steam pressure accompanying the varying demand on the boiler.

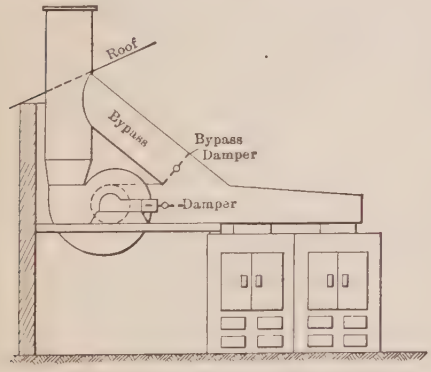


Fig. 382. — Induced Draft.

(**c**) Artificial draft is produced either by *steam jets* or by power-driven *fans*, and when developed by the latter it is generally called *mechanical draft.*

With *forced* draft (Fig. 381) the ash pit is "closed" (hermetically sealed) and the apparatus supplies it with air at a pressure above atmospheric (at a "plenum"); with *induced draft* (Fig. 382) the apparatus draws the gases from the boiler outlet, thus decreasing the pressure at that point below atmospheric; and with *balanced draft* these two systems are used in combination in a manner which will be discussed later.

(d) Fig. 383 shows one of the many forms of *steam jets* used for forcing the draft. Somewhat similar devices can be placed in the base of the stack (as is universally done in locomotives) to produce induced draft. Such apparatus is relatively low in first cost, but is very wasteful of steam, using generally not less than 5 to 8 per cent of the total steam generated, and it increases the stack loss because of the added moisture in the flue gas. Steam jets are convenient auxiliaries for meeting sudden or abnormal demands on the boilers, and the presence of the steam in the air supporting combustion tends to avoid the formation of clinkers. Fig. 384 shows a disc fan which is used in a similar manner.

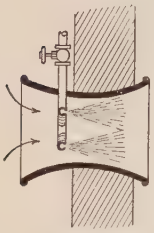


Fig. 383.

(e) With mechanical draft, the *fans* and their driving apparatus must be so designed as not to be affected by the dust, and with induced draft they must also be suitable for handling the hot gases without injury. If entire dependence is placed on fans for providing the draft, there should be duplicate (or auxiliary) apparatus installed to avoid plant shutdowns from failure of the draft apparatus. With a very short stack the fan equipment for forced draft costs roughly from 20 to 30 per cent as much as the equivalent brick chimney; while with induced draft the cost is about double that for forced draft as a larger fan ("exhauster") must be used because the gases are at high temperature. But though low in first cost, such apparatus depreciates rapidly, involves considerable expense for attention and maintenance, and uses for power from $1\frac{1}{2}$ to 5 per cent of the steam generated.

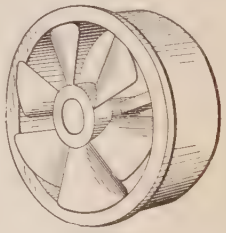


Fig. 384.

(f) With *forced draft* the gas pressure within the boiler setting is above atmospheric, hence the tendency for hot gases and flames to issue through cracks in the walls and also to belch forth upon the opening of the fire doors. To avoid the latter occurrence the blast is shut off, usually automatically, when the doors are opened.* The air should always be introduced into the ash pit in such manner as to subject the fuel bed to

* Some steamships using forced drafts have "closed firerooms" (stoke-holds) under pressure, and in such cases all leakage is into the interiors of the settings.

static pressure, or plenum, rather than to any localized blast action.

With *induced draft* (either natural or artificial) the pressure within the boiler setting is below atmospheric, hence there may be detrimental infiltration of cold air through cracks in the setting and through the fire doors when opened. With this system, however, the fuel bed burns more evenly, and demands less attention than in the other, and it is not necessary to shut off the draft before opening fire and ash doors. Usually a by-pass flue is provided (as in Fig. 382) so that natural draft alone can be used for light loads, or in case of accident to the apparatus.

With *balanced draft* the air is forced into the ash pit at sufficient pressure to become just atmospheric upon issuing from the surface of the fuel bed, and the gases are carried away from the combustion chamber by induced draft (either natural or artificial) of such intensity as not to cause a decrease of furnace pressure below atmospheric. The proper balance between the forced and induced draft is usually maintained by some automatic device which regulates the two systems simultaneously. With balanced draft (1) there is no tendency for leakage either into or from the furnace; (2) the fire is not affected by opening the furnace doors for adding coal or "working" the fire; (3) it is possible to burn the smaller sizes of fuel, which are otherwise worthless, and which must be burned at high rates of combustion but cannot be used with forced draft because of their fineness; and (4) very high rates of combustion can be used without detriment to economy.

CHAPTER XXXIII.

GAS PRODUCERS AND PRODUCER GAS.

281. Essentials of Producer-gas Apparatus. (a) Broadly speaking any apparatus in which gas is made is a "*gas producer*," but in engineering the term is almost exclusively applied to a class of apparatus producing gas largely by a process of partial or incomplete combustion. The gas made in such apparatus is known as "*producer gas.*"

(b) This gas has long been used for the heating of furnaces, the melting of metals, and a large number of similar purposes, but during the last twenty years it has come into particular prominence as a power gas, that is, a gas for use in internal combustion engines. It happens to be so constituted as to permit of high compression in the engine, thus giving high thermal efficiencies and, what is of greater importance industrially, it can be made at the point of consumption more or less easily and very cheaply as compared with most of the other combustible gases.

(c) Although the necessary apparatus differs considerably with the kind of fuel from which producer gas is to be made and with the purpose for which the gas is to be used, there are certain essential parts which generally exist in one form or another in all such apparatus. They are: (1) The fuel gasifier or "producer"; (2) some sort of "preheater" or "economizer"; (3) cleaning apparatus; and occasionally (4) a gas storage reservoir of some kind, large or small.

The first three parts are all shown in Figs. 5 and 391 to 394. In the particular types of plant shown in Figs. 5 and 391, the gas storage reservoir is practically nonexistent unless the pipe connecting the top of the scrubber with the engine cylinder be considered as partly serving that purpose.

282. Simple Theory of Producer Action. (a) As indicated above, the ideal producer makes gas by what is known as partial or incomplete combustion. In its simplest conception this

depends upon the combustion of carbon to carbon dioxide and then the reduction of this carbon dioxide to carbon monoxide by passing it over incandescent carbon. These reactions can be illustrated by means of Fig. 385.

(b) Assume the vessel there shown to be filled with a column of carbon, the lower part of which is heated to incandescence. If air enter at the bottom of this fuel bed, as indicated by the arrows, its oxygen will unite there with carbon to form carbon dioxide, according to the equation (see Eq. (342a))

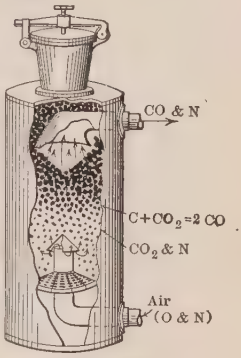


Fig. 385.

$$C + O_2 = CO_2 + 175{,}200 \text{ B.t.u.}, \quad (423)$$

which means that twelve pounds of carbon combine with thirty-two pou ds of oxygen to form 44 pounds of carbon dioxide and that (12 × 14,600 =) 175,200 B.t.u. are liberated per twelve pounds of carbon.

(c) This carbon dioxide would then be reduced to carbon monoxide while passing up through the incandescent carbon above, and the reaction would occur according to the equation

$$CO_2 + C = 2\,CO - 67{,}200 \text{ B.t.u.} \quad . \quad . \quad . \quad (424)$$

This means that the 44 pounds of carbon dioxide formed in the lower part of the fuel bed unite with twelve more pounds of carbon which will result in the formation of fifty-six pounds of carbon monoxide and the absorption of an amount of heat equal to 67,200 B.t.u., which quantity is easily obtained analytically in the manner described in the next paragraph.

(d) Imagine the process occurring in two steps: First assume that the forty-four pounds of carbon dioxide break up into twelve pounds of carbon and thirty-two of oxygen. This could only occur with the absorption of 175,200 B.t.u., equal to the quantity liberated when the combination took place. Then imagine the carbon and oxygen to combine with an additional twelve pounds of carbon to form the fifty-six pounds of carbon monoxide. This would be represented by (see Eq. (343a))

$$2\,C + O_2 = 2\,CO + 108{,}000 \text{ B.t.u.}, \quad . \quad . \quad (425)$$

which merely states that twenty-four pounds of carbon burning to carbon monoxide liberate (24 × 4500 =) 108,000 B.t.u.

The first process involved the absorption of 175,200 B.t.u. in breaking up CO_2, the second liberated 108,000 B.t.u. in the formation of CO, and the net result is the absorption of (175,200 − 108,000 =) 67,200 B.t.u., as given in Eq. (424).*

(e) The composition of the gas formed and the thermal efficiency of the process can now be determined: —

To produce the gas according to Eqs. (423) to (425), thirty-two pounds of oxygen are required per twenty-four pounds of carbon used and this oxygen will bring into the producer ($32 \times 77/23$ =) 107.1 pounds of nitrogen; hence the 163.1 pounds of gas leaving the producer will contain this weight of nitrogen in mixture with the fifty-six pounds of carbon monoxide resulting from the partial combustion, and will therefore have a composition of about 34.4 per cent CO and 65.6 per cent N by weight.

By volume the composition would be practically the same because the densities of CO and N are practically identical.

283. Efficiency, Simple Producer Action. (a) Had the twenty-four pounds of carbon used in Sect. 282 (c) been burned directly to carbon dioxide, they could have liberated $24 \times 14{,}600 = 350{,}000$ B.t.u. Burned to carbon monoxide they liberated only $24 \times 4500 = 108{,}000$ B.t.u. The difference,

$$350{,}000 - 108{,}000 = 242{,}000 \text{ B.t.u.},$$

must be the quantity of heat which can be produced by subsequently burning the carbon monoxide of the producer gas to carbon dioxide. This corresponds to 10,100 B.t.u. per pound of carbon.

(b) If the *thermal efficiency of the producer* be taken as the ratio of the heat which can be obtained by burning the cold gas to the heat which could have been obtained by burning the original carbon, it is in this case

$$\text{Ef}_c = \frac{\text{Calorific Value of Gas}}{\text{Calorific Value of Fuel}} = \frac{242{,}000}{350{,}000} = 69\%. \qquad (426)$$

Looked at in this way the process does not promise very well from a power-engineering standpoint. If the theoretical pro-

* It will be shown in a subsequent paragraph that this treatment does not tell the whole story, but for a first analysis it is accurate enough.

ducer-efficiency is only 69 per cent, the real efficiency could hardly be expected to be more than 50 to 60 per cent, and, with thermal efficiencies of internal combustion engines ranging from 20 to 30 per cent as an extreme value, the overall thermal efficiency of such a producer in combination with an engine would be low indeed. It will be shown later, however, that higher efficiencies are obtainable by modifying the process.

(c) The efficiency given above is what is called the *cold gas efficiency* and is really not the correct efficiency to use under all conditions. For power purposes the gas must be cooled approximately to room temperature before it can be advantageously used in an engine. This means removing all of the sensible heat given the material in the producer, and the cold gas efficiency is the proper value to use under such circumstances.

(d) The process as outlined results not only in the production of 163 pounds of gas, which can liberate 242,000 B.t.u. when burned, but also in the liberation of 108,000 B.t.u. in the producer. In any real case part of this latter heat will of course be used to supply unavoidable radiation and similar losses, but the rest will raise the temperature of the carbon and of the air fed to the producer and of the gas formed. Hence the gas would actually leave the producer with a very high temperature, about 2000° F. or more, and, by cooling it to room temperature, all of the heat liberated in the producer, and which was not lost by radiation or in other ways, could be obtained.

The temperature rise resulting from the liberation of a certain number of B.t.u. is equal to this number divided by the sum of the products of weight by specific heat of all the gases resulting from the combustion. The higher the temperature of the combustible gas and of the air before combustion, the higher will be the ultimate temperature attained.* Therefore, for furnace and similar work, where the object in burning the gas is to obtain high temperature, it is decidedly advantageous to have the apparatus located near the producer so that the sensible heat is not lost by radiation during transmission.

For such purposes the thermal efficiency of the producer is

* As the specific heats of gases increase comparatively rapidly at high temperatures, the temperature ultimately attained by any combustion will be lower than that given by the form of calculations suggested, as has already been shown. The error will be greater the higher the temperature attained.

correctly taken as the so-called *hot gas efficiency*, which is the quotient resulting when the sum of the total calorific value and the sensible heat of the gas leaving the producer is divided by the total calorific value of the fuel entering. Remembering that all heat which is liberated within the apparatus, and not lost by radiation and such, must be present in the gas leaving, the "hot gas efficiency" must be

$$Ef_h = \frac{\textit{Total Calorific Value of Gas} + (\textit{Heat Liberated in Producer} - \textit{Losses})}{\textit{Total Calorific Value of Fuel}}; \quad (427)$$

and if all the losses in the case previously considered be assumed at 20 per cent of the heat liberated in the producer, the hot gas efficiency for this case would be

$$Ef_h = \frac{242{,}000 + (108{,}000 - 0.2 \times 108{,}000)}{350{,}000}$$

$$= \frac{328{,}400}{350{,}000} = 93.5 \text{ per cent (approximately)},$$

a figure which is evidently much more promising than that previously obtained.

284. More Advanced Theory of Producer Action. (a) If the combustion processes indicated in the equations of the preceding section really occurred as there given it would be possible to pass a stream of carbon dioxide into one end of a tube containing hot carbon and have nothing but carbon monoxide issue from the other end. Experiment, however, shows that this is impossible, for, no matter what the conditions are, there will always be a certain amount of carbon dioxide mixed with the issuing carbon monoxide.

(b) Experiment further shows that, *other things being equal*, the higher the temperature in the tube the greater will be the proportion of carbon monoxide in the gas issuing, and the lower the temperature the greater will be the proportion of carbon dioxide.

(c) This is explained chemically by what is called "chemical equilibrium." Briefly, if no other variables need be considered, *at each given temperature, there are certain definite proportions of carbon monoxide and carbon dioxide which will be in equilibrium with carbon.* If a mixture of these gases in other proportions is

brought into contact with carbon, reactions will occur and continue until the equilibrium proportions corresponding to the given temperature are attained.*

(d) This equilibrium is well shown by the diagram of Fig. 386 which is plotted from experimental results obtained with carbon in a tube, as described in (a) of this section. In this figure the abscissas represent temperatures in Centigrade and Fahrenheit degrees and ordinates represent per cent of *CO* by volume. Subtracting these ordinates from 100 gives the percentages of CO_2, which are evidently shown to scale by distances from the curve to the 100 per cent line.

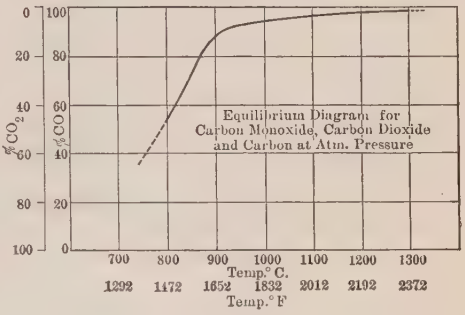


Fig. 386.

The curve shows that for low temperatures probably a very small amount of carbon monoxide would be found to be issuing from the tube in the experiment described above, while at high temperatures it shows the issuing gas to be composed almost entirely of carbon monoxide.

(e) In giving the effect of temperature on the composition as deduced from experiment, it was limited by the phrase "other things being equal." Experiment shows that the *pressure* at which the gases exist also has a certain effect upon the composition. The higher the pressure the greater the percentage of carbon dioxide in the equilibrium mixture at any temperature. Pressure variations are, however, so slight in producer work that their effect may be safely neglected.

(f) The *time of contact* is also of great importance. Chemical reactions do not occur instantaneously (that is, in time measured in infinitesimals) and the reactions in question, which lead to the equilibrium conditions plotted in Fig. 386, take a very appreciable time for completion. The higher the temperature the shorter the time necessary for the attainment of equilibrium conditions.

* Whether reaction then ceases, or whether counterbalancing reactions which do not further change the proportions of carbon monoxide and carbon dioxide continue, is a matter of indifference for the present discussion.

This is well shown in Fig. 387 which gives results obtained in experiments with carbon in the form of charcoal. In the figure each curve is an isothermal; that is, it shows the proportions of carbon monoxide and of carbon dioxide that will exist after gas, which was originally all carbon dioxide, has been in contact with carbon at a certain temperature for different lengths of time.

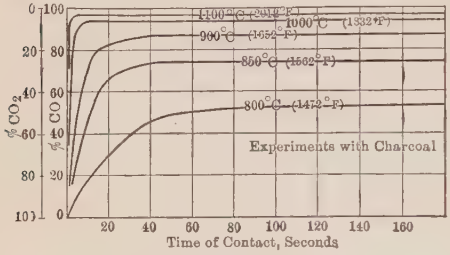


Fig. 387.

As before, the ordinates represent percentages of *CO* and the distances from the curve upward represent percentages of CO_2. It will be observed that, while it takes a time period of from 120 to 160 seconds to attain approximate equilibrium at 800°C. (as shown by the tendency of the curve to become horizontal at this point), it requires only 5 seconds to attain equilibrium with a very much higher percentage of *CO* at 1100° C.

(g) The effects of *both time and temperature* are well shown in Fig. 389, in which the three coördinates are time, temperature and volume per cent. of *CO*. The curves shown are those of Fig. 387, but here each curve is located in its own temperature plane. A surface can be imagined as passed through these curves and the coördinates of any point in it will show the relative percentages of *CO* and CO_2, which result at any temperature after any period of contact.

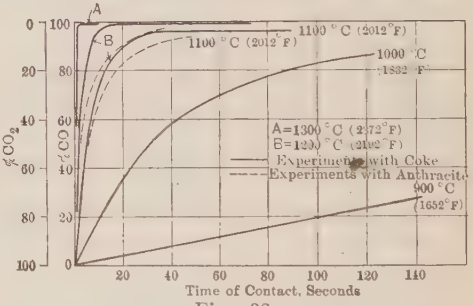


Fig. 388.

(h) Unfortunately the *surface condition* of the carbon has an effect upon the time required for the attainment of equilibrium. In general the more porous the carbon, and the smaller the lumps, the shorter will be the time required to attain the equilibrium corresponding to the given temperature. This is just what would be expected, as carbon of porous character and in

small lumps will expose most surface on which the reaction may occur.

The effect of surface (and possibly other) conditions is shown by a comparison of Figs. 387 and 388. The full lines in the latter represents the results of experiments made with carbon in the form of coke in lumps about the same size as those of the char-

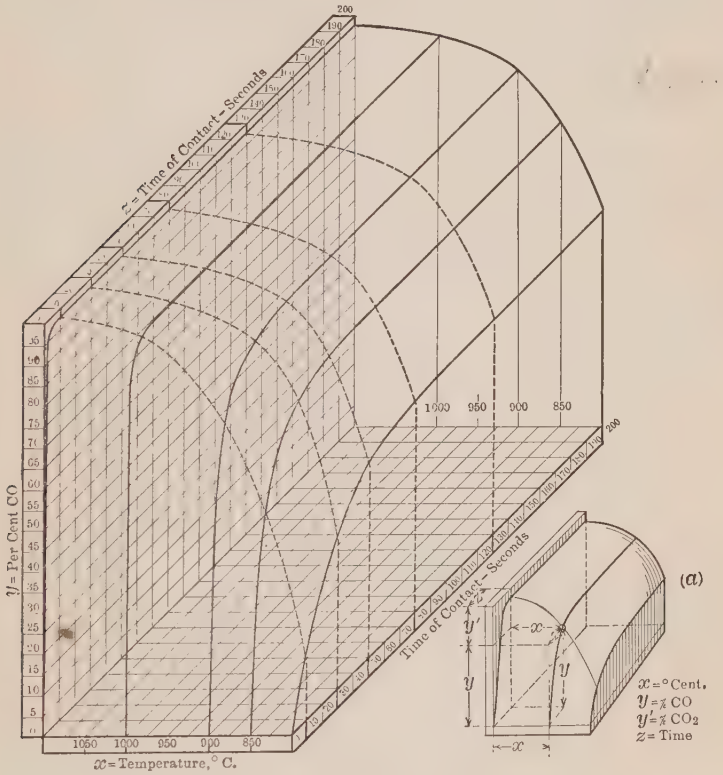


Fig. 389.

coal used in obtaining the results shown in Fig. 387. The dotted lines show similar curves for anthracite under approximately like conditions.

The curved surface shown in Fig. 389 is then only one of a number which differ in curvature with the character of the carbon. The more porous and the smaller the lumps the sharper will be the rise of the curves as they leave the temperature axis at the front, and the sooner will they become flatter as they recede.

(i) The preceding discussion is purely theoretical and leads to the following conclusions: For best producer operation (that is, the manufacture of gas containing the maximum amount of carbon monoxide and the minimum amount of carbon dioxide and nitrogen) the requirements are: —

(1) High temperature within the producer;
(2) Long time of contact between entering air, gas in process of formation, and hot carbon;
(3) Maximum porosity and minimum size of fuel;
(4) Theoretical air supply.

285. Practical Limitations. (a) In the real producer there are a number of practical considerations which materially modify the conclusions just given for the theoretical case. For instance, all real fuels contain ash and this will fuse and form clinker if the *temperature* becomes high enough.* Such clinker is very undesirable because it obstructs the gas passages between the lumps of fuel, making it difficult or impossible for gas to flow through certain areas. This results generally in more violent combustion in the parts of the bed which are still unobstructed, and this localized combustion materially augments the trouble by raising the temperature locally and causing the rapid formation of more clinker. The more or less complete obstruction of the gas passages will ultimately make continued operation impossible. Clinker also gives considerable trouble by fusing to the walls of the producer itself.

Thus, in actual operation, the fusing temperature of the ash sets the limit to the temperature allowable in the producer and this temperature varies considerably with different fuels; but, with those adapted to use in present-day producers, the temperature can generally be carried at such a value as to give a *theoretical* proportion of from 96 to 98 per cent of CO (with 4 to 2 per cent of CO_2) by volume in the issuing gas.

(b) *Caking* fuels also cause trouble in producer operation. The coalescence of the individual lumps decreases the percentage

* A number of experimenters are now operating producers at what are ordinarily considered exorbitantly high temperatures, by mixing with the coal some cheap material, such as limestone, which acts on the ash as a flux. The ash is thus made very fluid and is drained off just as it is in the case of blast furnaces. Several plants of this character are said to be in successful operation in Europe but they have not yet been commercially adopted in this country.

of voids in the fuel bed and thus obstructs the flow of gas. It also assists in causing "arching" so that the lower part of the fuel column may burn to ash and drop down while the upper part remains suspended above. Constant or intermittent stirring of the fuel bed (often combined with the maintenance of a fairly low temperature) are necessary with such fuels, although both stirring and low temperature have a detrimental effect upon the gas made. Stirring is often improperly done, and opens up fairly large free passages through the bed, thus allowing CO_2 and even air to pass through without coming into intimate contact with hot carbon.

(c) The theoretical requirement of long *time of contact* is more or less a relative consideration, as previously indicated; and the length of time needed was seen to depend both upon the temperature and upon the physical character of the fuel.

Remembering that a producer operates with a continuous flow of gas through the fuel bed, the time of contact between gas and carbon must be measured in the actual case by the length of time it takes a given particle of gas to pass through the fuel bed, and hence depends on the velocity of the gas passing through the producer and the length of the passage through the bed of fuel.

There is a practical limit to the *allowable depth of fuel bed* with any given fuel, in any given size, with any given type of producer. This limit is set by the *difference of pressure* necessary to cause flow through the bed. The length of the gas path through the producer being thus limited, the time of contact varies with the velocity which, in turn, depends on the diameter of the fuel bed. Large diameters will correspond to low velocities and long times of contact; small diameters will correspond to high velocities and short times of contact.

(d) This consideration would indicate a large *diameter of producer* to be desirable in every case, but there are two practical limitations which must be recognized: (1) The *cost* of the installation will increase as the size of the apparatus required per horse power increases; and (2) there will be difficulty in operating the producer under light loads, for when a producer which is of such diameter as to have a low gas velocity at full load is operated at a small fraction of that load, the small amount of air passing through may not be sufficient to keep the temperature

of the large bed of fuel up to that necessary for the formation of a high percentage of carbon monoxide.

The diameter of any given producer must therefore be a compromise between the large value desirable at full load and the smaller value which is desirable at light load and which involves less expenditure for equipment.

(e) Practice has shown that certain proportions are advisable with certain types of producers and certain kinds of fuel. In general it may be said that producers are built of such diameters that the amount of fuel gasified when carrying rated load is from 10 or 12 pounds per square foot of cross section of fuel bed per hour in the simpler types, up to 30 to 40 pounds per square foot per hour in the more complicated types of producers operating on particularly suitable fuels. The overload capacity is determined by the blast pressure available, by the clinkering temperature of the ash and the fusing or fluxing temperature of the producer lining.

(f) In the purchase of fuel, *porosity* can hardly be considered except in a general way, it being merely incidental to other considerations. The *size of lumps* can, however, be taken into account both as to effect on the operation of the producer and on the price, the smaller sizes generally costing less than the larger. The smaller the size of the fuel, with a given depth of column, the greater the difference of pressure required to pass the necessary volume of gas through the producer; and with a given maximum blast pressure, the lower is the capacity of a producer, if the same depth of column is maintained. Given a certain difference of pressure it is of course possible, by decreasing the length of path through that bed, to make any given quantity of air enter the bed per square foot of section in a given time; but this detrimentally shortens the time of contact or else necessitates an increase of diameter in order to reduce the velocity of flow to a satisfactory value.

Furthermore, with very small sizes the necessary velocity in a producer of a given diameter may be so great that parts of the fuel will be picked up by the gas and be carried out of the producer. This occurs to a greater or less extent with every producer in actual operation. Finely divided ash or a certain amount of finely divided or powdered fuel is practically always carried out by the issuing gas.

The larger the size of fuel the greater are the voids, hence the passages through the fuel bed are of greater cross section and the allowable velocity and blast pressure are less, but smaller surfaces are exposed for reaction.

(g) There are thus practical limits to the largest and smallest sizes of fuel which can be satisfactorily used in any given case. Sizes commonly used vary from about eight inches in diameter to pea anthracite. The larger sizes are generally mixed with smaller ones to decrease the passage areas in the fuel bed, while the smaller sizes very often have the finer particles screened out to increase the free areas.

(h) Lastly, practically no producer can be operated with the theoretical *air supply*. In order to supply the amount of oxygen necessary to produce the required quantity of CO (and the CO_2 which must necessarily accompany it), air in excess of the theoretical requirement must be passed through the producer. In general the smaller the size of fuel and the lower the velocity, the smaller need this excess be, but it can never be entirely eliminated. As a result, producer gas practically always contains more or less free oxygen and consequently an excessive amount of nitrogen. The typical analyses of producer gases in Table XXV show this.

286. Artificial Cooling of Producers (General). (a) In considering the difference between cold gas and hot gas efficiencies, it was shown that a large amount of heat in excess of that required to supply radiation and similar losses must be liberated within a producer by the very process to which the formation of producer gas is due. In actual producers using any of the ordinary fuels, the excess heat would quickly raise the temperature to a prohibitively high value. Clinker troubles would assume such magnitudes as to entirely prevent successful operation and in many cases there would even be danger of fusing the refractory lining of the producer shell.

(b) To prevent such difficulties producer operation is modified in several different ways by introducing some heat absorbing process to lower the operating temperature. Part of the excess heat is absorbed naturally to a certain extent with all real fuels, for they contain hydrocarbons and water which are vaporized (and to a certain extent modified chemically) at the

TABLE XXV. — TYPICAL ANALYSES OF PRODUCER GASES.*

Kind of fuel.	Constituents, per cent volume.								Calculated calorific value, B.t u /cu. ft. at 62° and 14 7 lbs.		Calorific value by test, B.t.u./cu. ft.		Remarks.
	CO_2.	CO.	H.	CH_4.	C_2H_4.	N.	O.	C_6H_6.	High.	Low.	High.	Low.	
Anthracite.....	3.90	27.30	12.30	1.00	....	55.50	....	....	138	132	147	....	
	6.40	22.00	8.20	2.40	....	61.00	....	....	123	113	122	116	
	3.60	20.00	5.30	3.00	0.20	67 50	0.40	....	114	106	118	113	
	10.50	19.50	7.90	0.70	....	61.40	....	....	95	90	102.6	....	
	6.03	22.38	13.38	1.96	0.36	55.65	0.803	....	138.4	125.7	....	....	Aver. of about 30 analyses.
Coke.....	4.80	27.60	7.00	2.00	....	58.60	....	....	130	125	140	....	
Bituminous....	....	27 0	12.0	....	....	57.0	0.3	....	126	120	140	....	
	5.7	22.0	10.5	2.60	0.6	58.2	0.4	....	135	126	150	....	
	9.48	18.28	12 9	3.12	0.18	55.6	0.04	....	133	122	152.1	....	
	14.50	12.00	29.0	2.00	....	42.5	....	....	153	136	154	....	
	9.12	17.54	11.73	4.28	0.36	57.24	0.265	....	137	125	....	....	Aver. of about 40 analyses.
Lignite.....	6.40	22.00	9.60	1.60	0.7	58.90	0.8	....	124	118	139	....	
	12.40	21 00	18.50	2.20	0.4	45 5	....	....	153	144	175	....	
	6.95	16.20	25.30	4.95	....	45.35	1.25	....	183	165	185	166.5	
	9.43	18.90	15.13	3.65	0.424	52 50	0.582	....	144	134	....	....	Aver. of about 10 analyses.
	12.40	21.00	18.50	2.20	0.40	45.50	....	....	153	141	175	....	
Peat.....	6.90	28 60	8.50	2.20	....	53 80	....	....	141	135	131	....	
Wood.....	21.30	9.86	54.14	3.45	....	10.53	0.26	0.46	247	217	246.9	....	
	13.90	20 03	21.00	2.79	0.60	41.80	0.185	Trace	137	128.7	....	....	Aver. of 5 analyses.
Oil.....	3.8	11.00	3.80	6.90	4.60	66.30	3.6	....	202	174	....	....	
	4.0	10.00	5.00	8.00	2.00	70 00	1.00	....	160	145	....	....	
	4.10	11.40	5.57	5.87	3.10	66.90	3.06	....	176	151	....	....	Aver. of about 10 analyses.

* From data collected by T. C. Ulbricht and C. E. Torrance. — Thesis "American Practice in the Rating of Internal Combustion Engines," Cornell Univ., 1912.

temperatures attained; but it is only with exceptionally wet fuels, such as poorly dried peat, that enough heat is absorbed in this manner to make operation practical. The reduction of temperature due to the presence of CO_2, nitrogen and moisture in the air (from the atmosphere) is also slight. Hence in actual operation it is necessary to have some artificial means of cooling.

(c) Two methods of artificially controlling the temperature are in use. They may be called

1. The "*Carbon Monoxide*" *Method*, in which burned producer gas is returned to the producer in mixture with the air supply, and absorbs heat largely by the reduction of contained CO_2 to CO; and

2. The *Water Vapor Method*, in which water vapor is mixed with the air supply and absorbs heat by reduction in contact with hot carbon.

The details of these two methods are considered in the following sections.

287. The "Carbon Monoxide" Method of Temperature Control. (a) Burned producer gas may be roughly said to consist of carbon dioxide and water vapor, mixed with nitrogen. Returning such material to the producer will effect cooling in two distinct ways: (1) The carbon dioxide passing over heated carbon will be more or less completely reduced to carbon monoxide; and (2) the water vapor in contact with hot carbon will be more or less completely broken up to form hydrogen, carbon monoxide and carbon dioxide.

(b) Under ordinary circumstances the first way will be the only one of appreciable magnitude because of the small amount of water vapor generally present when this method of cooling is used. The quantity of carbon dioxide which must be returned in order to maintain a given temperature in a theoretical case can be approximately determined if the amount of carbon dioxide and carbon monoxide which will be in equilibrium at that temperature and the amount of heat which must be absorbed to maintain the desired temperature are known. Knowing the amount of heat (see Eq. 424) which is absorbed for each unit weight of carbon present in carbon dioxide when the latter is reduced to monoxide, it is possible to determine how much carbon dioxide will have to be reduced and how much carbon

monoxide will result. It is then only necessary to find the amount of carbon dioxide which will be in equilibrium with this *CO*, to add this amount to that used in forming the monoxide, and the result is the total quantity of carbon dioxide which must be thus returned.

(**c**) It is interesting to note that nearly all carbon returned in the form of carbon dioxide is used repeatedly, being reduced to carbon monoxide in the producer, burned to dioxide in the apparatus utilizing the gas, and returned to the producer again for reduction, and so on. Hence the carbon furnished by the fuel will be less than the total carbon in the issuing gas by just the amount which is thus used over and over again. It should, however, be noted, that there is a natural limit to the amount that can thus be used. The carbon dioxide can be reduced to carbon monoxide only with the absorption of heat and this heat can come only from fuel carbon burned in the producer. The method is then simply one which results in the entrapping in available form of some of the heat which would otherwise be wasted. The actual amount of carbon which can thus be used repeatedly in any given case is comparatively small and the principal advantage of the process lies in the temperature control and uniform composition of gas (see (**d**) below) rather than in the saving of fuel.

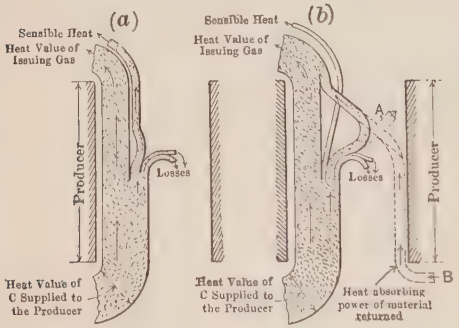


Fig. 390.

The operation of artificially cooled producers is shown diagrammatically in Fig. 390 (*b*) in comparison with the uncooled method of operation shown in Fig. 390 (*a*). The way in which more heat is made available at the expense of what would otherwise be lost is well shown by the width of the various streams.

(**d**) The fact that the time of ignition in internal combustion engines should ordinarily be varied with the composition of the fuel was mentioned in Sect. 212 (**h**). When gas is made in a producer controlled by the carbon monoxide method its composition is remarkably uniform from light loads to full loads, hence

the time of ignition can remain fixed without danger of large variations in thermal efficiency. This must be considered an advantage possessed by this process in comparison with that considered below.

288. The Water Vapor Method of Temperature Control. (a) Experiment shows that when steam is passed over incandescent carbon a certain amount of hydrogen is released, that the oxygen previously combined with it unites with some of the carbon to form carbon monoxide and carbon dioxide,* and that some of the steam will still remain unchanged regardless of the temperature attained and of the amount of carbon present. It is again a case of chemical equilibrium similar to that considered in Sect. 284 and the resultant composition will depend largely on the temperature.

No matter what the temperature is, a certain amount of heat will be absorbed in the process of decomposing the water and this is always greater than that liberated by the combination of liberated oxygen with carbon to form either CO or CO_2. Hence, the process, when used in a producer, must result in lowering the producer temperature, and by properly proportioning the amount of steam per pound of air the temperature can be directly controlled.

(b) The gas made by this process will contain hydrogen, and a small amount of methane as well as carbon monoxide, as a combustible constituent; and since all of the hydrogen and some of the carbon monoxide were formed in such a way as not to necessitate the introduction of nitrogen, there will be a smaller percentage of nitrogen in the resulting gas than when made by either of the processes previously described. The gas may, in fact, be considered as made by the theoretical process outlined in Sect. 282, with an admixture of hydrogen, carbon monoxide and a small amount of carbon dioxide, all resulting from the action of the steam.

(c) As a result of the presence of the hydrogen and of carbon monoxide not accompanied by its proportion of nitrogen, the calorific value of gas made by this process is higher than that made by those previously described.

(d) One of the disadvantages of this process is that the com-

* A small amount of methane, CH_4, is also found in all cases.

position of the gas is very changeable, the hydrogen content increasing from a very small amount at light loads to very large ones at heavy loads. This necessitates a constant shifting of the time of ignition if a uniformly high thermal efficiency is to be obtained, — the ignition occurring earliest with minimum hydrogen content. Such constant shifting with rapidly varying load is, however, not practicably attainable, hence the engine is apt to operate at widely varying efficiencies, and, in extreme cases, may not even operate satisfactorily.

(e) There are also many methods of applying this process which give poor results because of the method rather than the intrinsic nature of the process. Any method of controlling the steam supply which depends only on the instantaneous load on the engine must cause unsatisfactory operation in the following way: During a period of very light load the fuel bed has a tendency to cool down and if continued for any great length of time the temperature drop will be serious. Imagine full load to be demanded suddenly after such a period. The fuel bed temperature is hardly high enough to make the necessary quantity of monoxide, and if the producer is fitted with a device which will immediately throw on full steam supply with demand for full load, the fuel bed will be still further cooled by the deluge of water vapor. Many failures of otherwise successful producers have been due to just such actions.

Forms of steam control which depend upon the temperature of the gas issuing from the producer, or the equivalent of this, seem to give better results.

(f) The fact that water must be vaporized before it can be mixed with the entering air is often taken advantage of to conserve some of the sensible heat in the gas leaving the producer. An apparatus variously known as a *vaporizer*, an economizer, or by several other names, is so arranged that these gases, while passing through or around it, heat and vaporize water contained within it. Somewhat similar devices also called economizers, or, more correctly, *preheaters*, are often arranged to preheat the *air* on its way to the producer so that it may pick up more water vapor and also return to the producer some of the heat that would otherwise be wasted.

The upper part or cover of the producer shown in Fig. 5 forms a vaporizer, the vapor being picked up by the air supply as it

passes over the surface of the water. A somewhat similar device is shown in Fig. 393. Other forms, better known as economizers, are shown in Figs. 391 and 392.

289. Effects of Hydrocarbons in Fuels. (a) The behavior of real fuels in producers and the composition of the resulting gases are much modified by the presence of volatile hydrocarbons. These are distilled off within a producer and are very much modified by the high temperature of the heated fuel and refractory material before finally issuing with the gas.

(b) The tendency of all such mixtures of hydrocarbons when heated is to undergo changes, yielding carbon, hydrogen and new hydrocarbons, some of which are more volatile than the originals and others less volatile. If heating is continued long enough and at a sufficiently high temperature the ultimate products are practically hydrogen, methane and carbon (lampblack). The hydrogen and more volatile hydrocarbons, such as methane, form desirable constituents of the producer gas * and the carbon can be gasified if it remains in the producer, or it is comparatively easily separated if it passes out with the issuing gas. The less volatile hydrocarbons, however, if allowed to issue with the gas, will subsequently condense, giving a thick, viscous, or semi-solid material known either as *tar* or *pitch*, depending upon its composition and consistency. Such material is apt to cause pipe stoppages, to clog the gas cleaning apparatus, the engine valves and such.

(c) With *anthracite* fuels the amount of tar formed is comparatively small and gives little trouble as it is easily separated from the gas. *Bituminous* fuels, on the other hand, yield large quantities of tar if used in producers of the simpler kinds. Such tar must be separated from the gas if the latter is to be transported any distance from the producer, or is to be used in internal combustion engines or in any apparatus requiring it to flow through small orifices. This elimination not only entails the use of more or less costly apparatus, which generally consumes power, but also results in lowered thermal efficiency, as

* This statement is true as far as definite knowledge goes at present, but it seems probable that under certain conditions some of the products may prove undesirable because of chemical instability leading to spontaneous ignition at low compressions.

the calorific value of the separated tar represents, in many cases, a considerable portion of that of the original fuel.

(d) The most successful method of elimination so far produced depends upon the destruction of the tar within the producer. This is accomplished by passing the tar forming vapors (distilled off the freshly charged fuel), through an incandescent fuel bed before leaving the producer. The process taking place is called *cracking* and results in the formation of hydrogen, methane, small quantities of other very volatile hydrocarbons and solid carbon. The solid carbon largely remains within the

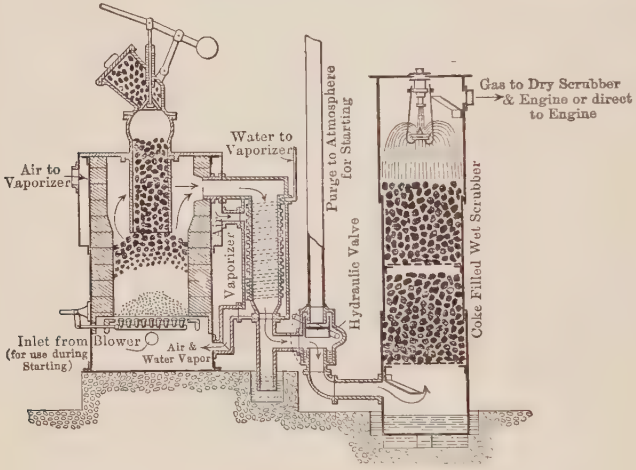


Fig. 391. — Up Draft Suction Producer.

producer bed and is subsequently gasified, while the hydrogen and other products of the cracked hydrocarbons pass off with the gas.

(e) This process can be carried on in an ordinary *up draft producer* operating much like that of Fig. 391, but modified so that, while the gas to be used is drawn off from the top of the main fuel column, the volatiles distilled off from the top of the freshly charged fuel in the extended hopper are piped around and introduced with the air entering at the bottom. This method has not met with great commercial success, although it is used to some extent in Europe for large installations.

(f) Another and a very successful method is to reverse the direction of flow of gas through the producer, introducing air

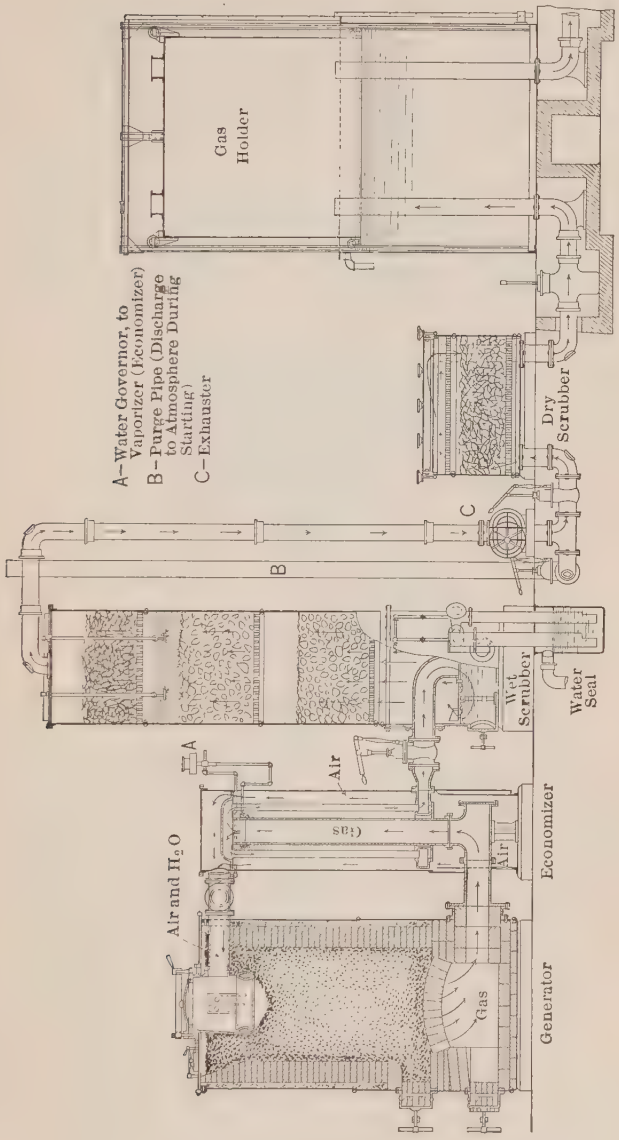


Fig. 392. — Down Draft Producer. (Loomis-Pettibone Type.)

and fuel at the top and removing gas at the bottom. This gives what is known as a *down draft producer*, one example of which is shown in Fig. 392. In this particular type, a bed of coke is ignited upon the brick arch which forms the grate, and the bituminous coal is fired upon this from above.

(g) Few of the down draft producers so far constructed have permitted of continuous operation because of the difficulty of removing the ash and clinkers without shutting down the producer. They are, therefore, generally operated intermittently, say for a week, after which they are cleaned out and restarted. The type shown in Fig. 393, which is known as a water bottom

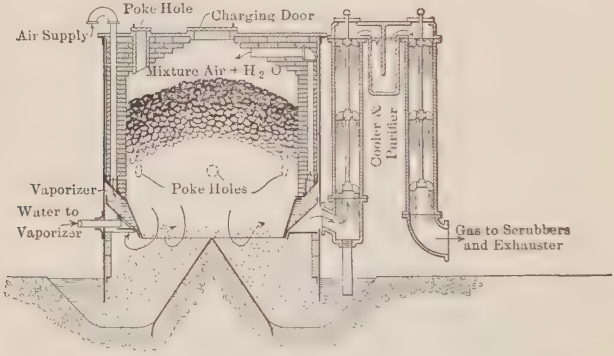


Fig. 393. — Down Draft Producer, Continuously Operated. (Akerlund Type.)

producer (see next section), overcomes this difficulty and permits of continuous operation.

(h) To circumvent the difficulties met in attempting to gasify bituminous and similar fuels in up draft producers, the so-called *double zone type of producer* is also used. One example of this type is shown in Fig. 394. This producer may be regarded as a down draft producer superimposed upon one of the up draft kind. Air enters both top and bottom and gas is drawn off near the middle of height. The only drawback is the necessity of carefully watching operations so that the upper incandescent zone may remain extensive enough to successfully crack the hydrocarbons and so that the combustion below the "gas offtake" may occur at just the proper rate to completely gasify all the coked material coming down from above.

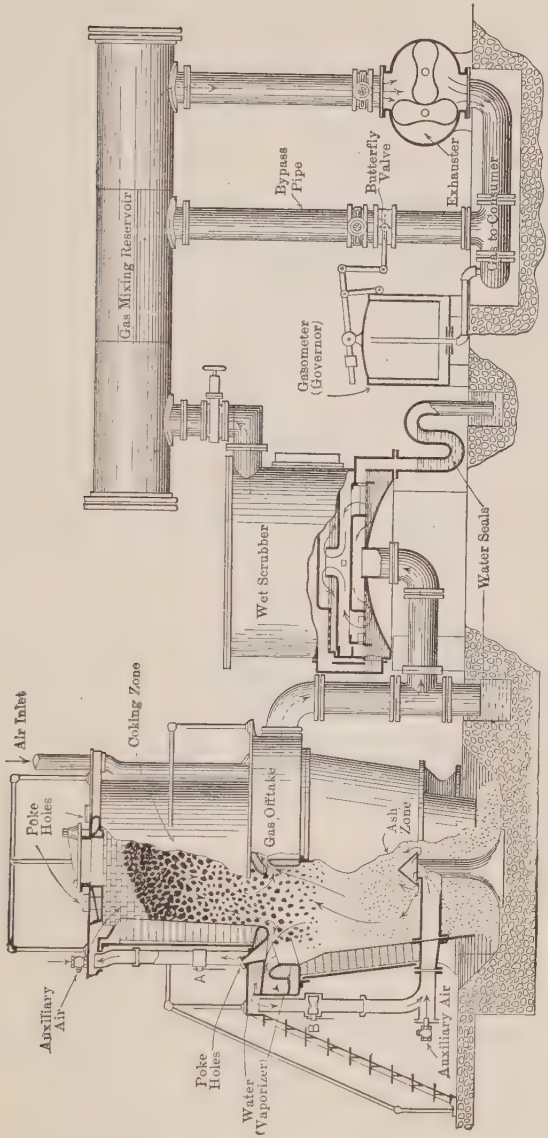
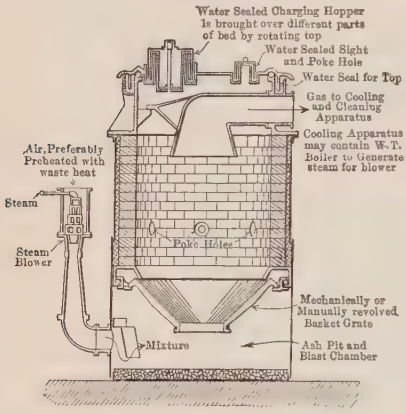


Fig. 394. — Double Zone Producer. (Westinghouse Type.)

290. Water Bottom and Grate Bottom Producers. (a) No matter what the type of producer, the column of fuel must be supported in some way. Producers are divided roughly into two types depending upon the way in which the fuel bed is supported. Producers arranged like those in Figs. 393 and 394, in which the bed of fuel is supported on a pile of its own ash, resting in a saucer shaped depression filled with water, are called *water bottom producers*. The shell of the producer must dip into the water by a sufficient amount to prevent the passage of air into the producer, or the escape of gas out of the producer, under the action of the greatest difference of pressure which will ever occur during operation.


Water Sealed Charging Hopper is brought over different parts of bed by rotating top
Water Sealed Sight and Poke Hole
Water Seal for Top
Gas to Cooling and Cleaning Apparatus
Cooling Apparatus may contain W.T. Boiler to Generate steam for blower
Air, Preferably Preheated with waste heat
Steam
Steam Blower
Poke Holes
Mechanically or Manually revolved Basket Grate
Mixture
Ash Pit and Blast Chamber
Fig. 395.

Producers of this kind possess the great advantage of permitting the convenient withdrawal of ash at any time during operation. They also dispense with almost all of the metal work found in other types at the bottom of the fuel column where the temperature is apt to become dangerously high if attendants are careless and where the rough work and sharp ash and clinker cause rapid depreciation.

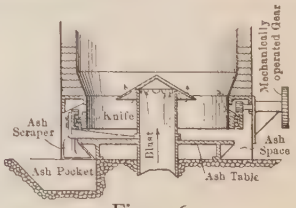


Fig. 396.

(b) Producers in which the column of fuel is not supported by ashes in a water sealed saucer may be roughly grouped under the head of *grate bottom producers*. Examples are shown in Figs. 391, 392, 395 and 396. The grates may be of any degree of complexity from the simple grid of *cast iron bars* or of plain *iron pipes*, or the *arch of fire brick* shown in Fig. 392, to the most complicated of mechanical grates, such as the *rotating and scraping devices* shown in Figs. 395 and 396, or a *rocking grate* much like that used under steam boilers, as shown in Fig. 391.

(c) Mechanical grates, operated continuously by power in

large sizes and intermittently by hand in small ones, are decidedly advantageous. They make possible the easy working down of ash and clinker, which in other cases would have to be barred down by poking from above, from the sides and through the bottom. When continuously in motion they tend to maintain uniform conditions within the producer, shaking the fuel column sufficiently to keep it open, to work down ash and to break up clinker.

Combined with a depth of ash sufficient to seal against leakage in or out, or with a water bottom, they afford ideal operating conditions.

291. Induced Draft and Forced Draft. (a) In developing the theory of the producer it was assumed that air could be made to enter and the resultant gas to leave. This flow can only be produced by maintaining a difference of pressure between inlet and outlet orifices. Two distinct methods are used for creating flow in this way. In one the pressure on the entering side is raised above that of the atmosphere; that is, air is pumped into the producer. Such producers, which are operated under what corresponds to *forced draft* in boiler practice, are called *pressure producers*. The pressure of air and gas within the producer is greater than the atmospheric pressure outside of the producer shell by the amount necessary to cause flow through the producer and all subsequent apparatus. One great disadvantage of such types is the fact that a leak anywhere in the apparatus results in the outflow of poisonous producer gas. Opening of poke holes for inspection of the fire or for stirring up the bed will result similarly. For such reasons pressure producers must always be operated in well ventilated structures, preferably without side walls where climatic conditions permit.

Air is generally pumped into such producers by a *steam jet blower* similar to that shown in Fig. 395. With proper proportions the amount of steam can be regulated so as to just equal that required for cooling the producer by decomposing in contact with the hot carbon. In general this steam must be under so high a pressure that a separate boiler is necessary for its generation, but in large plants this is not a great disadvantage. In some types the steam is generated in a vertical water tube boiler which receives the hot gas coming from the producer. In this

way part of the sensible heat in the gas is returned to the fuel column.

Any other form of air pump can be used; a very common one is that shown at the right in Fig. 394. As shown it is used as an *exhauster*, but an exactly similar device can be used as a *blower*, the only difference being in its location and method of connection.

(b) The other method of operation referred to above causes flow through the producer and subsequent apparatus by lowering the pressure at the outlet to a value below that of the surrounding atmosphere. Atmospheric pressure at the inlet is then sufficient to cause flow. Such producers, operated under conditions corresponding to induced draft in boiler practice, are known either as *induced draft producers* or as *suction producers*, depending upon the apparatus used for reducing the pressure.

When an exhauster like that in Fig. 394, or any similar apparatus, is used the producer is called an induced draft apparatus; when a gas engine operated by the producer draws its own charge through the system by lowering the pressure during each suction stroke as shown in Fig. 391, the apparatus is called a suction producer.

(c) One great advantage of all induced or suction draft systems is that any leak always results in the flow of air *into* the apparatus rather than escape of gas out of the apparatus. Such air may, in extreme cases, furnish oxygen sufficient to burn an appreciable quantity of the gas within the apparatus, as for instance in case the leak is immediately above the fuel bed in the producer where the gases still have a high enough temperature to ignite, and this would result in a diminished output of power gas, but could not ordinarily endanger human life. A leak at a point beyond the producer would result in the mixture of air and cold gas which could be entirely counterbalanced by the admission of less air to the apparatus in which the gas is subsequently burned. However, such mixtures of air and gas within the apparatus represent a possible source of trouble as they may sometimes acquire explosive proportions and there is always the possibility of ignition. The high pressures resulting from explosions would endanger the apparatus and possibly human life, but can easily be guarded against by providing some form of pressure relief such as a water seal or large flat door, or plate, with minimum

inertia so as to permit of rapid opening with minimum pressure rise.

(d) There are a few types of producer plants so constructed that they operate on what is known as a *balanced draft.* This is generally achieved by using the equivalents of one blower and one exhauster. The combined action of the two is such that the pressure within the producer itself is not greatly different from atmospheric, that on the outlet side being generally maintained at a value equal to atmospheric. The dangers associated with leakage in or out are thus minimized.

292. Mechanical Charging. (a) Most of the producers used in power plants are charged by hand, particularly in the smaller sizes. With small producers the shell is generally made of such depth that it will hold sufficient fuel for from three to six hours' operation without recharging. Such producers are commonly charged when starting in the morning, again at about midday and finally at night before shutting down.

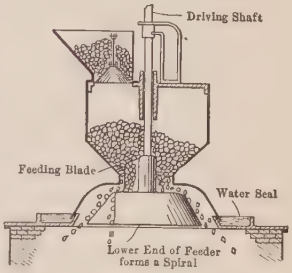


Fig. 397.

(b) With large producers it is generally found best to charge at shorter intervals because more uniform results are thus obtained and because the fuel column is more easily handled when this is done. Where frequent charging is necessary a mechanical device such as is shown in Fig. 397 has many advantages, the most prominent of which are:—

(1) Uniform rate of charging so that there are no sudden fluctuations in quality of gas as happens when large quantities of green fuel are charged at long intervals;

(2) Uniform distribution of fuel over the entire diameter of producer, — a very difficult matter in hand charging of large producers unless much hand leveling is done, which generally permits the admission of considerable quantities of air or the discharge of large volumes of gas during the operation; and

(3) Saving of labor and hence of operating expense.

293. Cleaning Apparatus. (a) The gas leaving a producer has a comparatively high temperature and carries in suspension

more or less solid matter and also vapors which, upon cooling, will condense to form water and tar. The function of the cleaning apparatus is to cool the gases and to remove solids, water and tar.

(b) In the broad sense every part of the apparatus beyond the producer outlet flange is cooling and cleaning apparatus, but commercially the term is applied to the several distinct units such as *wet* and *dry scrubbers*, *tar extractors* and such.

(c) The *methods employed for cooling* are almost obvious from the figures of actual producers shown. A certain amount of sensible heat is removed from the gas by air or water, or both, in the economizer or its equivalent. Part of the sensible heat of the gas is lost by radiation from the pipes connecting the various parts. By far the largest amount is generally removed in wet scrubbers, by bubbling the gas through water, or by passing it through a space filled with a very fine spray of water, or over water films on coke or on similar solid material, or by a combination of these methods.

(d) The *methods of removing solids* (and in some cases liquids) depend upon three principles: (1) Separation by gravitation, (2) separation by change of direction, and (3) separation by wetting solid particles and retention of such wetted material.

Settling of solid (and liquid) matter will result to a certain extent when the velocity of flow is sufficiently reduced. This reduction in flow may be brought about either by an enlargement in the size of the passage or by decrease in volume due to lowering the temperature of the gas, or by both these processes. In fact more or less separation of the kind always occurs in the pipes because of natural cooling. Separation by change in direction occurs whenever the gas passes through an elbow or similar fitting, the solids (and liquids) having a tendency to travel to the outside of the curve.

Special apparatus for utilizing these first two principles is seldom fitted to producers, and particularly not to those used for power work, though the fact that such separation must always occur to a certain extent as the gas flows through the passages may be utilized in design to lighten the work required of the following apparatus. It should also be taken into account in designing the piping by arranging openings through which cleaning can be easily effected.

By far the largest amount of solid material is removed by wet scrubbing which is also used for the cooling effect. The apparatus is so constructed that the solid particles are well wetted and are then allowed to separate out by gravity, or they are "scrubbed" out by bringing them in contact with wetted surfaces to which they adhere. Separation by gravity is well shown in the cases where gas is bubbled through water, the particles of dust on the surface of the "bubble" being wetted and caught by the liquid, after which they slowly settle to form a sort of mud.

Separation by wet scrubbing is well illustrated by the operation of coke or grid filled towers or "scrubbers" such as those shown in Figs. 391 and 392.

Even the best wet scrubbers will allow a small amount of dust to pass through them. This material, together with mechanically entrained water and some tar, is often finally separated in a "*dry scrubber*" filled with excelsior or sawdust, such as that shown in Fig. 392.

After the tar-forming vapors have been condensed, the small particles or "droplets" of tar behave in much the same way as do the dust particles. The apparatus used for dust removal, and in particular the wet scrubber, removes large proportions of tar as well.

(e) When the tar content of the gas is great, as when bituminous coals are used in a producer which does not provide for breaking up the hydrocarbons within the fuel bed, it is often necessary to use a separate **"tar remover."** These are generally mechanically operated scrubbing devices in which the gas is well wetted by a spray of water and then brought into forcible contact with moving and stationary surfaces. The tar collects upon these surfaces and the liquid is driven off or drained off continuously. Such separators often take forms resembling fan blowers, or series of propellers or impeller wheels, with adjacent units rotating in opposite directions.

294. Producer Gas from Oil. (a) Many attempts have been made to construct producer-gas plants which would successfully gasify crude oil and fuel oil, but most have resulted in failure. There are, however, a few plants, of several different types, in successful operation, which indicates that the problem of gasifying oil in a producer is not impossible of solution.

(b) The difficulties met in attempts to gasify oil are similar to those experienced with the gasification of the volatiles in bituminous coals and similar fuels. Either tar, or lampblack, is generally produced in large quantity, and gives trouble in cleaning, besides reducing the efficiency.

(c) One of the solutions of this problem is notable for its simplicity. The producer is arranged for down draft and is built with a brick arch grate similar to that shown in Fig. 392. A bed of incandescent coke is maintained on this arch and the oil is sprayed into the upper part of the producer, the resulting gas passing downward through the coke bed. All tar-forming vapors are destroyed by cracking and the resultant lampblack is nearly all caught in the coke bed which is thus automatically replenished.

CHAPTER XXXIV.

UTILIZATION OF WASTE HEAT—FINANCIAL CONSIDERATIONS.

295. General. (a) It has been seen that in connection with steam power plants very large amounts of heat are wasted in the flue gas (loss *c* in Fig. 3) and in the exhaust steam (width *E* in Fig. 3). The profitable reduction of these losses is obviously of the greatest importance, and it is the object of this chapter to outline briefly the different methods of its accomplishment and some of the more important problems connected therewith. In most cases it will be seen that some of the waste heat is used in increasing the temperature (sensible heat) of the feed water, thus reducing the amount of fuel required to convert the feed water into steam.

296. Utilization of the Heat in the Flue Gases. (a) One very common method of saving some of the heat that would ordinarily be wasted up the stack is to heat feed water by passing it through tubes which are surrounded by the flue gases after they have left the boiler. The heating apparatus in this case is commonly called an "*Economizer*." Its use effects a saving of heat which in exceptional cases may amount to as much as 15 per cent of the total calorific value of the fuel. The apparatus, its method of operation, advantages and disadvantages, etc., will be discussed in detail later.

(b) In certain instances, some of the heat of the flue gas can be used profitably for heating the air used in the furnace; and if the local conditions are favorable, the hot gases may be used in drying-kilns and such.

But in all cases where the temperature of the flue gas is decreased there is a detrimental effect on the draft (if natural), to offset which entails an additional expense for an increased height of stack, or for artificial draft apparatus and its operation.

297. Utilization of the Heat in the Exhaust Steam. (a) It has been seen that lowering the pressure of the exhaust steam issuing from a prime mover results in an increase of the available percentage of the total heat furnished by the boiler and hence reduces the proportion wasted in the exhaust. This decrease of pressure is commonly effected by using a *condenser*, of which there are many types. But there is in each power plant a limit of vacuum beyond which it does not pay to go; and even where the best vacuums are used, the exhaust steam still contains the larger part of the heat that is brought from the boiler, and this heat is nearly all surrendered to the condensing water. However, some of this heat may be returned to the boiler with the feed water (this being saved) but the proportion is generally quite small if the vacuum is good. In Fig. 3 this return is shown by the lower stream line, for one particular arrangement of plant. Condensers and methods of supplying the feed water with heat from the exhaust steam will be discussed more in detail later.

(b) When the steam is exhausted at atmospheric pressure, the feed water can be heated nearly to 212° by it and thus quite a considerable saving may be accomplished; but only a small percentage of the total exhaust steam of the entire plant can be profitably utilized in this manner. Frequently the main prime mover is operated condensing and the auxiliary apparatus non-condensing, the exhaust steam of the latter being used for feed-water heating. This results, in most instances, in more profit than arises from operating the auxiliaries condensing. The pieces of apparatus in which the feed water is heated by the exhaust steam are called *feed-water heaters;* they will be discussed in detail later.

(c) When the local conditions are suitable, some of the heat of the exhaust steam can be used in *industrial processes* which require temperatures lower than that corresponding to the exhaust pressure. Thus, for example, a steam prime mover might furnish power for an industry in which the heat of the exhaust steam could be utilized in dryers, or in kettles used for digesting various materials, and the condensate, with its sensible heat, might be returned as feed water to the boiler. In other industries in which solutions, having temperatures of vaporization below 212°, are evaporated in "*evaporating pans*" at atmospheric

pressure, or in "*vacuum pans*" under partial vacuums, the latent heat of the exhaust steam can be used to supply the heat necessary to evaporate the water from the solution; and in such cases not only may the vacuum pan act as a condenser for the power plant and thus reduce the back pressure on the prime mover, but the hot condensate may be returned as boiler feed.

(d) Many plants are situated in localities where the artificial *heating of buildings* is necessary for a large portion of the year. For such heating, low-pressure steam, i.e., steam at or near atmospheric pressure, is satisfactory; hence, the exhaust steam from engines suitably located can often be used for heating purposes for many months in each year.

As in the case of vacuum pans, the heating system can sometimes act as a condenser for the power plant, but in such cases the vacuum carried (if any) is very imperfect, the pressure not being much below atmospheric. Such "*vacuum systems*" are generally operated at a pressure of but 1 or 2 pounds below atmospheric, despite the fact that the lower the pressure of condensation, the greater is the latent heat surrendered by the steam. The reasons for not using greater vacuums are: (1) Lowering the temperature of the steam in the radiator necessitates a greater amount of radiating surface (which involves greater first cost), and (2) lowering the pressure makes it more difficult to keep the joints tight (that is, to prevent the inflow of air), for even if the heating system is of only moderate extent, there are hundreds of joints and it is difficult to insure permanent tightness in all of them.

In other heating systems, called *Pressure Systems*, the steam is at a pressure somewhat above atmospheric, the back pressure on the engine being generally from 5 to 20 pounds gauge pressure.

(e) Supposing that the condensate from the heating system is used for feed water (without loss of temperature) and that all the heat in the exhaust steam above feed-water temperature can be used for heating purposes, then, in the ideal case, the efficiency of the combined system is 100 per cent, for all of the heat which is given to the steam by the boiler and not converted into useful work is utilized in heating. In the actual case some of the heat is lost by cylinder radiation, by mechanical friction of engine and driving machinery, and by useless radiation in

pipes between engine and heating system; but even then the efficiency of the combination is relatively high while it is in operation. In fact, the cylinder radiation and dissipation of heat due to friction and work may not be waste if the same amount of heat would otherwise have to be used for warming the engine room.

Whether or not it would pay financially to utilize the exhaust steam for heating buildings depends on the location of the power plant, the length of the annual period of time during which the heating is necessary, the percentage of the total steam that can be used during such periods, the excess cost of equipment over that otherwise required and many other items which need not be considered in this brief discussion.

298. Heat Transmission. — In order to transfer the heat from the hot gases, or steam, to the feed water or other absorbing media, some kind of heat transmission must occur. Hence to properly understand the operation of condensers, economizers, feed-water heaters and similar apparatus, one must have a knowledge of the general theory of the transmission of heat. The subject will, therefore, be discussed (in the next chapter) before such apparatus is considered in detail.

299. Financial Considerations. (a) Suppose the installation of certain apparatus would effect a substantial saving in the weight of coal used; then from the standpoint of *heat utilization* there would be a gain. But suppose, further, that the expense chargeable against the installation and the operation of the apparatus itself would be more than the saving in the cost of coal; then, of course, the installation would not be profitable *financially.*

Obviously the advisability of the adoption of additional equipment depends on whether it will effect, in the long run, a saving greater than all expenses in any way chargeable against it.

(b) The capital invested in apparatus must be guarded against fall in value in order to protect the investor; but the apparatus is subject to decrease in value because of wear and possible accidents, and also because it may become obsolete by the introduction of improvements. This decrease in value is called "*depreciation.*" Therefore, each year there must be set aside a certain

sum (*a*) so that the amount thus accumulated plus the remaining market value of the apparatus will at least equal the investment. The more rapidly the apparatus deteriorates or becomes obsolete the greater is the annual depreciation to be set aside.

Furthermore, the capital must receive yearly *interest* (*b*) to be profitable; and as the investment increases so also do the expenditures for *taxes* (*c*), and *insurance* (*d*); and should additional space be demanded by the apparatus, there may be increased annual *rent* (*e*) to pay. These items, and perhaps some others, constitute what are called the *Fixed Charges* against the apparatus.

(c) Besides these items, the yearly *cost of operating* the apparatus must be considered, the principal items of such additional expense being some or all of the following: — (1) *Labor or attendance;* (2) *fuel consumption;* (3) *water used;* (4) *oil, waste and other supplies;* (5) *repairs and maintenance*, and possibly other items.

If the saving in expenditure for fuel per year should be greater than the sum of items (*a*) to (*e*) and of (1) to (5) and of any others not included, the installation of the apparatus will be a source of profit, otherwise not.

(d) It is not within the scope of this book to enter into the detailed discussion of the financial problems connected with power-plant engineering; but it is deemed necessary to show that the heat saving is not the final criterion. The foregoing very brief discussion is given for that purpose and to make clear to the reader what is meant when such phrases as "aside from the financial considerations involved " are used in the chapters which follow.

CHAPTER XXXV.

HEAT TRANSFER.

300. General. (a) In previous chapters it has been assumed possible to transfer heat from body to body at will, limited only by the law that a body cannot gain heat from one at a lower temperature unless energy is expended to cause the transfer. It is now necessary to investigate more closely the phenomena connected with the "flow" of heat under the "driving force" of a temperature difference.

(b) At the outset it must be clearly understood that from the engineer's viewpoint the whole subject of heat transfer is in a most undeveloped state. Many experiments have been made, numerous laws have been suggested, and much that is true has been recorded; but there are still many points about the subject which are matters of dispute and the settlement of which is anxiously awaited. It will be shown later that this is not so much due to the lack of scientific knowledge as to the lack of means of applying known facts, and of inability to analyze the exact conditions under which the heat transfers occur.

301. Heat Conduction. (a) Assume the metallic bar shown in Fig. 399 to be so insulated along its entire length that no heat can be dissipated by it to the surrounding atmosphere. Assume further that the ends are so arranged that heat can be continuously supplied to the bar at end A and the same amount continuously removed from end B. Under these circumstances the heat supplied will all flow along the length of the bar, i.e., flow through the bar. Experience shows that under such conditions the temperature at B will always be lower than the temperature at A, that is, that there must be a temperature difference if heat is flowing.

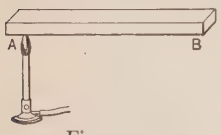


Fig. 399.

This is very similar to the phenomena met in the flow of electric currents in similar conductors. It is necessary that a difference

of potential exist between two points, A and B, if an electric current is to flow between them. In the one case then electricity flows "because" of a difference of electrical potential or electromotive force, in the other heat flows "because" of a difference of temperature, or, paralleling the above, a difference of "heat potential."

(b) Since it is supposed that the molecules of a substance move faster when at a high temperature than when at a low one, the sensible heat associated with the substance may therefore be conceived as being measured by the intensity of molecular motion, and heat conduction may be considered as merely the imparting of such motion to successive groups of molecules along the path of heat flow. According to this view when one end of a solid body is heated the molecules begin to vibrate more and more rapidly but they impart some of their energy to those molecules immediately adjacent to them, and these in turn pass on some to their neighbors, and so on through the entire substance.

(c) The laws governing this sort of heat flow are comparatively simple. To develop them assume the two parallel planes A and B, in the conducting body shown in Fig. 400, to each have unit area, to be unit distance apart and to be maintained at temperatures T_a and T_b, the former being one degree Fahrenheit greater than the latter. Then there will be a flow of heat from A to B; and, assuming no loss from the walls of the intervening body, heat will have to be supplied at A and removed at B at exactly the same rate as it flows between these points, if these temperatures and the flow are to be maintained constant.

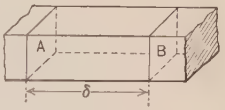


Fig. 400.

(d) Experiment shows that under such conditions a very definite amount of heat will flow from A to B per unit of time in any given material and this quantity is called the *Specific Heat Conductivity*. It will hereafter be designated by the Greek letter α,* and, as used in most engineering calculations, it is the number of B.t.u. flowing per *hour* in the material between two parallel planes with area of each equal to one square *foot*, with

* The letter λ is very commonly used for this, but because it has already been employed in this book to represent another equally important quantity it is thought best to prevent confusion by adopting the above unusual symbol.

one *inch* space between and with a difference of temperature of *one Fahrenheit degree.**

(e) It is easy to see that if planes having areas of twice this amount, i.e., two square feet, are assumed, twice as much heat would flow between them in a given time. The heat flow, or conductivity, therefore varies directly as the cross section, in the same way that electrical conductivity does.

Similarly, to cause heat in quantity α to flow between two planes of unit area, but at *two* units distance apart, will require twice the temperature difference that is needed when they are but one unit apart. This can easily be seen by imagining an intermediate plane at unit distance from each of the others. One degree of temperature difference will cause α heat units to flow from plane A to the intermediate one and plane B must be one degree lower than the intermediate to maintain the same rate of flow. In other words heat conductivity, like electrical conductivity, varies inversely as the length of the conductor.

(f) If ΔQ = the heat flow between two parallel planes of equal area in a given material in one hour,
θ = the temperature difference in Fahrenheit degrees,
S = area of each of the two planes in square feet, and
δ = distance apart of the planes in the conductor, in inches,

then, from the two statements in (e) above,

$$\Delta Q = \theta \cdot \frac{S\alpha}{\delta} \text{ B.t.u.} \dagger \quad . \quad . \quad . \quad . \quad . \quad (428)$$

And, if $S\alpha/\delta$ be called the conductivity of the heat path, its reciprocal $\delta/S\alpha$ may be called the *heat resistance*, R, just as the reciprocal of electrical conductivity is called the electrical resistance. If this is done equation (428) may be written

$$\Delta Q = \theta \div (\delta/S\alpha) = \theta/R,$$

the last term of the expression resembling Ohm's Law, but giving the heat flowing in unit time in terms of temperature difference divided by heat resistance, instead of electrical flow in unit time in terms of voltage difference divided by electrical resistance.

* This curious mixture of units is of convenience in engineering calculations.

† For flow between inner and outer surfaces of a cyclindrical wall see Appendix.

This form of expression can be used to find heat flow with any complicated combination of resistances just as is done in electrical problems, and, in general, paths with resistances in parallel or in series might be considered. However, in practical cases resistances in series are generally the only ones of importance, and for such instances

$$\Delta Q = \theta \div (\Sigma R). \quad . \quad . \quad . \quad . \quad . \quad . \quad (429)$$

(g) The specific heat-conductivity, α, varies, in general, with the kind of conductor in about the same way as does electrical conductivity; thus, good conductors of electricity are generally good conductors of heat, and vice versa. It also varies with purity of material, being different for instance for pure copper and copper containing small quantities of other metals, and with temperature much as does electrical conductivity. The conductivity α_t at any temperature above, or below, a chosen datum can be expressed in terms of the conductivity α_0 at datum temperature by the equation

$$\alpha_t = \alpha_0 (1 + \beta t);$$

and with 32° F. as datum this becomes

$$\alpha_t = \alpha_{32} \{1 + \beta (t - 32)\}, \quad . \quad . \quad . \quad . \quad (430)$$

in which β is a constant which has values varying with the material, being positive with some and negative with others.

The values of α_{32} and β are given in Table XXVI for some of the common heat conductors used by the engineer.

Comparison of the specific conductivities tabulated will show that for metals they are several hundred times as great as for water, and that for this latter substance the conductivity is several times the value for gases. Stagnant gases are about the poorest conductors, and stagnant water is nearly as bad.

302. Heat Transfer by Convection. (a) When *fluids* (liquids and gases) have their temperatures raised locally, the heat energy is distributed through the mass of fluid not only by conduction, such as was just considered, but also by what is known as "convection." Most liquids are comparatively poor heat conductors and practically all gases are very bad ones, but under proper conditions heat may be transferred to distant parts of the fluid very quickly by convection.

TABLE XXVI.* — SPECIFIC CONDUCTIVITY OF VARIOUS MATERIALS.

Materials.	α_{32} †.	β†.	
Cast iron	330.00	−0.0004	Average values for gray iron. Variations with composition very great.
Wrought iron (unworked)	450.00	−0.0009	See next below as indication of variation.
Wrought iron (worked)	240.00	−0.0006	
Steel (soft)	300.00	−0.0003	See below as indication of variations.
Steel (mod. hard)	240.00	−0.0003	
Steel (very hard)	180	.....	
Copper (pure)	2400	−0.0002	Values given by different experimenters vary considerably. Probably due to variations in purity and condition.
Copper (commercial)	2100	+0.0001	
Brass (yellow)	420	+0.0014	Varies greatly with composition.
Brass (red)	540	+0.0008	
Aluminum (pure)	750	+0.0003	
Aluminum with { 0.5% Fe, 0.4% Cu }	1041	+0.0002	
Cylinder oil	0.784	−0.0015	Naturally varies with kind of oil, cylinder oil not being a definite compound.
Water	2.6158	+0.0053	These values seem best authenticated. Authorities differ greatly.
Air	0.11989	+0.0017	Varies with humidity, etc.
Hydrogen	0.71933	.....	
Fire brick	.....	.....	6.948 at 1300° F. Varies considerably with composition of brick.
"Insulating" materials	0.4 to 1.2	.....	Such materials as cork, cellular paper, asbestos mixtures, etc.

* Compiled largely from the Landolt-Bornstein-Myerhoffer Tabellen, and from "Hutte," Des Ingenieurs Taschenbuch.

There is still considerable uncertainty and disagreement regarding the specific conductivities of the various substances and the U. S. Bureau of Standards is now carrying on investigations on this subject. The results will presumably be published eventually in a bulletin.

† α is heat in B. t. u. conducted per square foot, per degree difference, per hour, per inch thickness of material. β is the constant in Eq. (430).

(b) Practically all fluids increase in volume when heated, that is, their density decreases. Local heating will therefore cause local decrease of density; but this will disturb the mechanical equilibrium of the fluid and there will be a tendency for the heated portions to rise. This will be more marked the more intense and local the heating, and it results in the flow of the heated material through the rest, that is, currents are formed, or "*circulation*" occurs. This process very rapidly distributes heat energy to all parts of the mass even though the fluid be a poor conductor of heat. Examples of convection currents caused by local heating are shown by the arrows in Figs. 401 and 350.

Fig. 401.

(c) The marked distinction between heat conduction and heat convection can now be clearly shown, if the views expressed are assumed correct: Heat conduction is due to the individual motions of single molecules, while heat convection is the common transportation of groups of molecules.

(d) No attempt will be made to give an expression for the rate at which heat is distributed through a fluid by convection, as it would be very complicated and of little use at best. It would at least involve differences of temperatures and densities, specific heats, viscosity and molecular friction. In general it may be said the heat transfer by convection will increase with temperature difference, or with the intensity of local heating, and will be greater the less the viscosity of the material.

303. Heat Transfer by Radiation. (a) Experiment shows that bodies at all temperatures radiate energy at the expense of their associated heat, which energy, when stopped or absorbed by another body or medium, becomes evident as heat energy. This does not mean that the radiated energy is in the form of heat when on the way between the two bodies; in fact, if heat energy is to be considered as connected with the motion or condition of molecules, radiant energy of this kind cannot be heat as it will pass through a vacuum devoid of molecules of any kind.

(b) Like light, radiant energy is supposed to be transmitted by the hypothetical "ether," and to be a vibratory form of energy. It is further commonly supposed that the molecules of a body start such vibrations in the ether at the expense of

part of their energy, and that the energy associated with molecules of other bodies can be augmented at the expense of these ether vibrations. Whether the ether exists or not, and whether the process goes on in this way or not, is really immaterial. The facts remain that a body can lose heat by radiating energy, which is not what is commonly called heat after leaving that body, and that substances can be raised in temperature, vaporized and so on, by receiving such radiated energy. This energy will hereafter be called *radiant energy.**

(c) The rate at which heat energy is radiated by a body increases very rapidly as the absolute temperature is raised. Unfortunately the exact law governing has not yet been definitely determined, but it seems probable that the amount of energy radiated varies with the fourth power of the absolute temperature. The heat ΔQ_R radiated per unit of surface per unit of time by a body maintained at a constant *absolute* temperature T is then given by the equation

$$\Delta Q_R = kT^4, \quad . \quad . \quad . \quad . \quad . \quad . \quad (431)$$

in which k is a constant, which depends on the character of the material.

The net loss of heat from the body by radiation is not given by this equation however. As any radiating body must be surrounded by others with definite temperatures, it must be receiving radiant energy as well as sending it; hence, the net result of such an interchange would be a loss or gain of heat equal to the difference between that sent and that received. On this basis the net heat lost per unit of time by unit surface of a body maintained at temperature T_1 (abs.), radiating to another parallel surface maintained at lower temperature T_2 (abs.), and with *vacuous space* between the two surfaces, would be

$$\Delta Q_{RN} = kT_1^4 - kT_2^4 = k\,(T_1^4 - T_2^4). \quad . \quad . \quad (432)$$

which is known as *Stefan's Law.*

(d) Since the radiant energy, like light, travels or radiates in all directions from the surface of the body which serves as its source, equations like those just given must be used with a certain

* The name "*radiant heat*" is often given to what is here called radiant energy. It is not adopted in this book because of the confusion of ideas which may result from its use; see (i) of this section.

amount of care. The radiating and receiving surfaces may be so arranged that all energy lost by one is received by the other (in which case Eq. (432) applies), or they may be so arranged that part of the energy is not caught (and Eq. (432) should then be modified).

In Fig. 402 the hot surface under consideration is supposed to be a small area S in the plane ab, which is of infinite area and has the same temperature throughout. The plane AB is a similar one of infinite extent having a uniform but lower temperature. It is obvious that the solid angle ϕ, representing the extreme angle with which the rays from surface S strike plane AB, approaches 180° as a limit, and that all energy radiated from S must be intercepted by AB or pass through it. In such case the hot surface S is said to "see" nothing but the cold surface and the radiant energy received from S by the cold surface is given in Eq. (431).

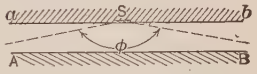


Fig. 402.

If, however, the surfaces are arranged as in Fig. 403, in which the cold surface is again represented by AB, it is evident that the solid angle ϕ is considerably less than 180°. All rays from S passing outside of this angle miss the cold surface entirely and are lost in the space beyond. The part of the total radiant energy intercepted by the cold surface would then be equal to that given by Eq. (431) multiplied by the ratio of the solid angle ϕ to the solid angle 180°. Note, however, that this does not give the net heat lost by the *hot* surface, for this will be all that can be lost through the solid angle 180° minus all that is gained through that same angle. Other cases can be analyzed in similar manner, the amount of radiant energy received by a body depending on the angle ϕ.

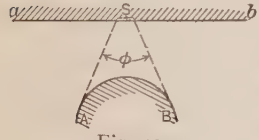


Fig. 403.

(e) The condition of the *surface* of a body determines to a considerable extent the rate at which it will give off or absorb radiant energy. Dull black surfaces are excellent radiators and absorbers. Polished metallic surfaces are very poor in both respects.

(f) Some few substances are practically transparent to radiant energy, that is, they allow practically all of it to pass through their structure without absorption, but all absorb more or less.

Every substance will absorb radiant energy with the same wave lengths as that which it radiates, and the theoretical limit of transparency to radiant energy would be attained with a body which radiated energy of one wave length only and hence absorbed radiant energy of that wave length only.

Most solid substances radiate energy of many different wave lengths and absorb in an equally broad fashion. Gases on the other hand radiate energy of only one or very few wave lengths and are proportionately transparent to radiant energy.

(g) In the case of two dull black, parallel surfaces of the same material and with vacuous space between, Eq. (432) will give the approximate net number of B.t.u. of radiant energy interchanged per hour per square foot of surface, if k has the value of about 16×10^{-10}, the temperatures being on the Fahrenheit scale.*

No real body has exactly the properties of the ideal black one, but sooted and lamp-blackened surfaces generally approach the ideal case within 5 per cent or less.

(h) In connection with Eq. (432) it should be noted that even if the two radiating surfaces are so arranged that each "sees" only the other, kT_1^4 will represent all the heat lost by body 1, but kT_2^4 will not necessarily represent all lost by body 2, by radiation in the direction toward body 1; this would be true only if the vibrations caused by both parallel surfaces were exactly alike.

(i) In connection with the subject of heat radiation it may be well to call attention to an anomalous expression in common engineering usage. All apparatus which is maintained at a temperature higher than that of the surrounding atmosphere loses heat to the latter and this loss is commonly spoken of as heat lost by "radiation." As a matter of fact only part of it is lost in such manner, the major portion being dissipated by convection, and a smaller part by conduction through the atmosphere.

304. Heat Transfer in Engineering Apparatus. (a) The three distinct methods of heat transfer so far considered are never

* For detailed discussion of the subject see Bull. 2, U. S. Bureau of Standards, 1905; pg. 107 of Bull. 18, U. S. Bureau of Mines; and Dalby, Heat Transmission, British Inst. of M. E., 1909, the latter containing references to over 500 papers on the general subject of heat transmission.

really found existing separately in any actual engineering problem, for, in general, all three methods of transfer are operating at the same time. Nor does the engineer as a rule have to deal with heat transfer in or through but one substance, or from a single simple substance to another single simple substance. In general, his problems are so complicated that in the end it is found simpler, in the present state of knowledge, to design by the use of empirical or semi-empirical equations rather than to attempt a rational treatment of each case.

(**b**) An idea of the sort of problems which occur can be given by considering a single case analogous to practice and developing the ideal equations for it, in so far as this can be done.

Imagine, for example, a sheet of metal separating two mediums at different temperatures, as is shown semidiagrammatically in Fig. 404, in which B is a section through the metal perpendicular to its surfaces bb' and cc', while A and C represent sections through the mediums on each side of the plate. The dotted lines aa' and dd' represent isothermal planes in these mediums, the material in plane aa' having a temperature t_1, and that in plane dd' having a lower temperature t_2.

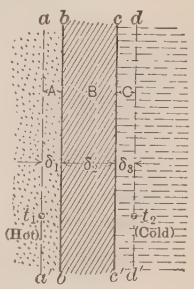


Fig. 404.

From what has already been said about conduction, it is evident that heat will flow, or be conducted, from the plane aa' through the mediums A, B and C to the plane dd' so long as the temperature difference is maintained. If the only method of heat transfer be assumed to be conduction, the heat flow can be calculated by Eqs. (428) and (429).

(**c**) The problem of conduction may be considered to be the determination of the amount of heat which can be made to flow by the temperature difference $(t_1 - t_2)$, through the three prisms with lengths δ_1, δ_2 and δ_3 inches, arranged in series as shown in perspective in Fig. 405 (a), in which the planes aa' and dd' are similar to those in Fig. 404.

(**d**) Investigating now in detail the assumed problem of heat conduction in connection with Fig. 405 (a), it is evident that there must be a constant drop of temperature along the length δ_1, if heat is flowing along this first prism. This is shown graphically by the line from t_1 to t_b at (b) in the figure, the temperatures being

represented by ordinates above an arbitrary chosen line which is not shown. The temperature drops steadily from a value t_1 at the plane aa' to a value t_b at the surface bb'.

At this surface there is an abrupt and marked temperature drop to t_b' which is necessary to overcome the surface resistance and make the heat enter the second medium, for careful experiment shows that a surface offers a certain resistance to heat flow, as it is found that a temperature drop must occur at a surface to cause heat to enter any given material. This so-called *surface or contact resistance* is often compared with that offered by a joint in an electrical circuit. But while there are points of resemblance, there are also many differences between the two cases, hence the parallel should not be carried too far.

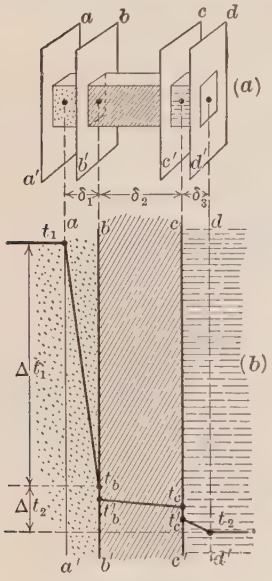


Fig. 405.

There is then a steady drop throughout the length δ_2 of the second prism until the temperature t_c is reached, the line from t_b' to t_c, in general, having a different slope from the line $t_1 t_b$ because of difference in the specific conductivity. At the surface cc' there is again an abrupt drop from t_c to t_c' and then the temperature decreases through the third medium until the assumed temperature t_2 is reached in the plane dd'.

In the example illustrated in the figure, gas and water are the mediums on the opposite sides of the plate, as is the case with boiler heating surface. But in the boiler there are additional resistances due to the soot on the external surfaces and scale and grease on the interior walls.

(e) Since heat flow is equal to temperature difference divided by resistance, the amount of heat per hour could be found in this case if all the resistances were known. The resistances being in series they are additive, as previously indicated in Eq. (429), and therefore in this case

$$\Delta Q = \frac{\theta}{R_1 + R' + R_2 + R'' + R_3}, \quad \cdot \cdot \cdot \quad (433)$$

in which R_1, R_2 and R_3 are the resistances of the paths δ_1, δ_2 and δ_3, and R' and R'' are the contact resistances of the planes bb' and cc' respectively. Then, remembering that *unit* cross section has been assumed, and that $R = \delta/S\alpha$, this equation may be written

$$\Delta Q = \frac{\theta}{\frac{\delta_1}{\alpha_1} + \frac{1}{\alpha'} + \frac{\delta_2}{\alpha_2} + \frac{1}{\alpha''} + \frac{\delta_3}{\alpha_3}}, \quad . \quad . \quad . \quad (434)$$

in which subscripts are used as in Eq. (433) and the symbols have the same meaning as in Eq. (428) excepting that no idea of length is attached to the *specific* heat-conductivities α' and α''.

Evidently the *total conductivity* per unit area is the reciprocal of the denominator in Eq. (434). If this is represented by K, then for any area S, the equation becomes

$$\Delta Q = KS\theta. \quad . \quad . \quad . \quad . \quad . \quad . \quad (435)$$

(**f**) Such equations as Eq. (434) or (435) can of course be solved for any given case if the values of the specific conductivities (α) or the total conductivity (K) are known; but in any real case such a calculation would be of little value as heat transfer will also be produced simultaneously by radiation and convection, the latter generally being forced to a certain extent. So great is the effect of convection in most engineering problems that it is often the most important consideration as can be well shown numerically. If heat is transferred from a metal plate to quiescent water under such conditions that convection currents are practically eliminated, the amount of heat transferred per square foot per hour per Fahrenheit degree difference of temperature will be of the order 2.8 B.t.u. ($= \alpha_t$). If on the other hand the water be in violent motion, or in ebullition so as to assist convection as much as possible, the heat transferred may be of the order 1500 B.t.u.

(**g**) It has already been seen that stagnant gases and water have conductivities several hundred times poorer than metals. Hence the *stagnant film* of fluid that adheres to the surfaces of the metal plates increases greatly the difficulty of heat transmission to and from such plates. These surface films may be regarded as being constituted of the molecules of fluid caught in the microscopic irregularities of the plate's surface, or of those which they entangle or retard. They act as heat insulators

which prevent the hotter particles of one fluid, and the colder particles of the other, from coming in contact with the plate. Obviously violent agitation of the fluids tends to destroy or reduce the thickness of the films and thus makes the conditions more favorable for heat transmission. Therefore the more violent the circulation or convection currents (within reason) the more rapid will be the heat transfer per square foot of surface; and this is not only because of the effect on the film, but also because the hotter portions of the fluid are brought to the plate at a more rapid rate. If gas and water are the two fluids, as in the steam boiler, the temperature drop Δt_1 (in Fig. 405 (*b*)) necessary to pass the heat through the gas film is relatively very much greater than the drop Δt_2 through the metal of the plate and through the water film; in some cases it may represent 98 per cent of the difference between the temperatures of the hot gas and the water.

The important part played by such films can be shown by an example: If a bunsen flame is placed below a metallic vessel containing boiling water the flame will not quite touch the metal but will spread out into a sheet at a distance of about $\frac{1}{30}$ to $\frac{1}{40}$ of an inch from the plate. Because of its high conductivity the plate on the gas side can be only a few degrees higher in temperature than the water, hence through the very short distance of $\frac{1}{30}$ to $\frac{1}{40}$ of an inch there must be a drop of temperature from that of a bunsen flame to a value only slightly above that of water boiling at atmospheric pressure.

(h) Some of the difficulties which arise in actual engineering problems involving the transmission of heat, and the reason for using empirical or semi-empirical formulas, will now be apparent. These problems are still further complicated by the coatings of scale, grease, soot, paint and other material on the surfaces of the transmission plates and by the relative directions of the flow of the fluids on the opposite sides of the plates. The effect of this flow can be analyzed quite accurately, as will be seen in the succeeding sections. It will first be discussed in a general way and later the mathematical treatment will be given in more detail.

305. Effectiveness of Heat Transmitting Surfaces. (a) It has been seen that the rate of transmission of heat through a plate depends directly on the difference between the tempera-

tures of its two surfaces. Obviously when the temperatures of the fluids on either side of the plate are maintained constant, the temperature drop is the same at all points over the surface, hence the rate of transmission and effectiveness of surfaces is uniform over the whole area. But when there is flow of one or both of the fluids, the conditions are quite different.

(b) Imagine, for instance, that the tube in Fig. 406 is surrounded with boiling water (temperature constant) and that hot gas flows from a to b, becoming cooler as it progresses. Then the average temperature drop (difference of temperature) (θ_{m_1}) through the wall back of surface S_1 is greater than that (θ_{m_2}) at surface S_2, and this latter is greater than that (θ_{m_3}) at surface S_3 and so on through the length of the tube. With a tube of infinite length the gas could theoretically be cooled to the temperature of the water and the temperature difference at the end b would be zero. Thus $\theta_{m_1} > \theta_{m_2} > \theta_{m_3} > \ldots \theta_{m_n}$ and each portion of the surface is less effective than those preceding and more so than those following. The nearer the temperature of the gas approaches that of the water the less effective is the adjacent heating surface, although it costs as much per square foot as the more effective portions. Hence, as was seen in Sect. 261(c), there is in each case some particular extent of surface which will give the greatest financial return.

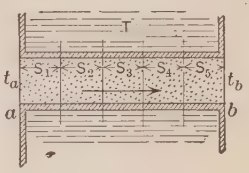


Fig. 406.

Obviously as the curve of temperature change of the gases is not straight, the mean temperature difference for the surface as a whole is not one-half the sum of the initial and final differences. Before the case can be analyzed mathematically it will be necessary to find the true value of the mean temperature difference.

(c) Besides the foregoing case, the cold fluid may flow and the hot one may be at constant temperature, or both the cold and hot fluids may be flowing and the currents may be either in the same direction or in opposite directions. In each of these additional cases the temperature difference varies over the surfaces, but the methods of variation are quite different from the case described in the preceding paragraph. All these cases will be considered in detail in later sections.

(d) In all cases of heat transmission through plates from hot

fluids to cold ones it may be noted that, neglecting radiation losses, the heat surrendered by the hot fluid must equal that received by the cold one and must also equal that flowing through the intervening material; hence

$$C_h W_h (T_a - T_b) = \Delta Q = C_c W_c (t_b - t_a), \quad . \quad . \quad (436)$$

in which

ΔQ = heat transmitted in a unit of time.
C_h and C_c = the specific heats of the hot and cold fluids.
W_h and W_c = weights of the fluids flowing per unit of time.
t_a and t_b = temperatures of cold fluid at ends a and b (Fig. 406).
T_a and T_b = temperatures of hot fluid at ends a and b (measured on the same temperature scale as that used for t_a and t_b).*

If the object is to have the cold fluid abstract a certain *quantity* of heat ΔQ in a given time with initial temperature t_a, it may be accomplished with large weight W_c of material leaving at low temperature t_b, or by a small weight leaving at high temperature, and similarly in regard to the quantity of heat supplied by the hot fluid. Obviously the final *temperature* attained by either fluid may be controlled by regulating the weight of material flowing per unit of time.

(e) Again, in all cases of heat transmission (neglecting radiation losses), the heat given up by one medium and received by the other must equal the conductivity of the path multiplied by the area of surface transmitting heat and by the temperature difference, as in Eq. (435). However, in case one or both fluids flow, the temperature difference is not constant but, as has just been seen, varies from point to point, hence a mean temperature difference θ_m must be used. For all conditions, then,

$$\Delta Q = KS\theta_m, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (437)$$

in which

ΔQ = heat transmitted (B.t.u. per hour)
= heat lost by hotter medium
= heat gained by cooler medium.

* T is not here used to represent absolute temperature but merely that of the hotter medium measured on the same temperature scale as t. Its use avoids the employment of primes, additional subscripts, or other complications that would be necessary to distinguish between the temperatures of the hot and cold bodies if the same letter, such as t, were used for both.

K = conductivity of heat path (B.t.u. per sq. ft., per hr., per ° F.).
S = total surface (sq. ft.).
θ_m = mean temperature difference (° F.)
= θ when no flow occurs.

(f) But before Eq. 437 can be used θ_m must first be determined. As will be shown later it is given for all cases by the equation

$$\theta_m = \frac{\theta_a - \theta_b}{log_e (\theta_a/\theta_b)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (438)$$

in which

θ_a = temperature difference at end a of the surface.
θ_b = temperature difference at end b of the surface.

Therefore, no matter what the conditions of flow,

$$\Delta Q = KS \frac{\theta_a - \theta_b}{log_e (\theta_a/\theta_b)}, \quad . \quad . \quad . \quad . \quad . \quad (439)$$

or

$$S = \Delta Q \div K \frac{\theta_a - \theta_b}{\log_e (\theta_a/\theta_b)} \quad . \quad . \quad . \quad . \quad . \quad (440)$$

306. Cases of Heat Transmission through Plates. (a) There are five cases of heat transmission through plates, and Eqs. (436) to (440) apply to all of them. They will be described briefly in this section and in more detail later.

(b) **Case I.** (T = *const.*) *A hot substance at constant temperature T surrenders heat to a flowing cold substance, whose temperature t is increased.* Surface condensers and feed water heaters are examples of this case, for in both of these heat of the exhaust steam (at constant temperature) is surrendered to water which is raised in temperature as it flows through the apparatus. This case is shown by the curves in Fig. 407, in which ordinates are temperatures and abscissas are extent of surface. The upper curve is for the hot fluid and the lower for the cold one, the flow being toward the right. It will be seen that the final temperature of the cold body depends on the total length of surface, and that, as the flow progresses, the temperature t of the cold medium gradually approaches that (T = const.) of the hot fluid, and the temperature difference and value of the surface (per square foot) becomes less.

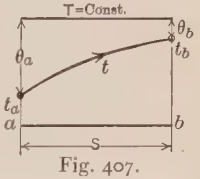


Fig. 407.

The efficiency of the heating surface is evidently

$$\text{Ef.} = \text{Heat transmitted} \div \text{maximum amount absorbable}$$

$$= \frac{C_c W_c (\theta_a - \theta_b)}{C_c W_c \theta_a} = \frac{\theta_a - \theta_b}{\theta_a} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (441)$$

(c) **Case II.** (t = *const.*) *A substance at constant temperature* (t) *receives heat from a hotter flowing substance whose temperature* (T) *decreases.* An example of this is the steam boiler, in which the boiling water (at constant temperature t) receives heat from the hot gases which decrease in temperature T as they progress. This case is the reverse of Case I, and Eq. (441) applies except that $C_h W_h$ would be substituted for $C_c W_c$. Fig. 408 is the diagram for this case.

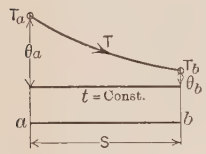


Fig. 408.

(d) **Case III.** *Parallel Flow.* The term is understood by engineers to mean parallel flow in the *same* direction on opposite sides of the plate. *The hot and cold substances both flow in the same direction, their temperatures converging nearer to equality as they progress,* as shown in Fig. 409 in which the arrows show the direction of flow.

With finite surface the heat transmitted is $C_h W_h (T_a - T_b) = C_c W_c (t_b - t_a)$. If the object is to absorb heat, the maximum amount which the cold fluid could receive is $C_c W_c \theta_a$, and the Comparative Efficiency (to be used in comparisons with other cases) is therefore

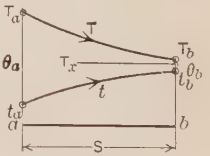


Fig. 409.

$$CEf_c = \frac{C_c W_c (t_b - t_a)}{C_c W_c \theta_a} = \frac{t_b - t_a}{\theta_a} \quad . \quad . \quad (442)$$

But if the object is to cool the hot fluid the maximum amount of heat that could be surrendered is $C_h W_h \theta_a$ and in that case the Comparative Efficiency is

$$CEf_h = \frac{C_h W_h (T_a - T_b)}{C_h W_h \theta_a} = \frac{T_a - T_b}{\theta_a} \quad . \quad . \quad . \quad (443)$$

No matter how extensive the surface, T_x, in the figure, is the limit of temperature to which the hot fluid can be cooled and the cold one heated. The heat available for transmission is $C_h W_h \theta_a$ (or $C_c W_c \theta_a$) and with infinite surface only the part

$C_hW_h\,(T_a - T_x)$, or $C_cW_c\,(T_x - t_a)$, could be transmitted. Hence the maximum possible efficiency is

$$Ef_h = (T_a - T_x) \div \theta_a, \quad . \quad . \quad . \quad . \quad . \quad (444)$$

or

$$Ef_c = (T_x - t_a) \div \theta_a, \quad . \quad . \quad . \quad . \quad . \quad (445)$$

which is less than that attainable in any of the other cases. It would therefore appear that this arrangement should always be avoided; however, if only a relatively small portion of the available temperature head θ_a is to be utilized, parallel flow may be advantageously used, as under these conditions it requires less heating surface (and hence the initial cost is less) to produce the same result than is required in some other arrangements.

(e) **Case IV.** *Counter flow. The hot and cold substances flow in opposite directions, the temperature of the former approaching the lowest temperature of the latter, and vice versa, as they proceed.* This is shown in Fig. 410, in which the directions of flow are shown by the arrows, T_a being the initial temperature of the hot fluid and t_b being that of the cold one. The relation between the heat absorbing, or surrendering, capacities of the two fluids is given by the ratio of C_cW_c to C_hW_h and this determines whether the two curves diverge or converge or are parallel. With *infinite* surface it can be shown that the hot medium would be cooled to the initial temperature t_b of the cold one if $C_hW_h < C_cW_c$; or the cold medium will be raised to the initial temperature T_a of the hot fluid if $C_cW_c < C_hW_h$; and in this ideal case the efficiency is unity, since all the heat possible is transmitted with the materials in question. Hence the only limit to the efficiency is the extent of the heating surface. But a comparison of Fig. 410 with Fig. 409 will show that the mean temperature head is less with counter current flow than with parallel flow, hence to accomplish the same degree of heating (or cooling), and to obtain the same efficiency, more extensive surface (at greater cost) is required. Counter current apparatus is therefore characterized by high possible efficiency and by economy of amount of heating or cooling material, as an offset for the greater surface necessary.

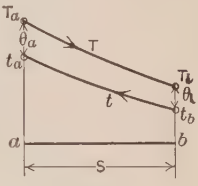


Fig. 410.*

(f) **Case V.** (*T and t both constant.*) *Transmission of heat from a hot fluid of constant temperature (T) to a cold one which*

* Fig. 410 is for $C_cW_c > C_hW_h$. If $C_cW_c < C_hW_h$ then $\theta_b > \theta_a$.

remains at constant temperature (t). The vacuum pan is an example of this case, since the latent heat of the steam (at constant temperature) furnishes the heat for evaporating (at constant temperature) the other medium.

(g) The mathematical treatment of these five cases will now be discussed in the succeeding sections.*

307. Case I. (T = Const.) A Hot Substance at Constant Temperature Surrenders Heat to a Cold Fluid which Flows. (a) Assume the conditions shown in Fig. 411 and imagine that W_c pounds of the cooler material flow over the small surface δS in a given time and that as a result their temperature is raised an amount $\delta t\ (=\delta\theta)$. Then, if the specific heat of the cooler material be C_c, the heat δQ absorbed *from* the area δS per unit of time is

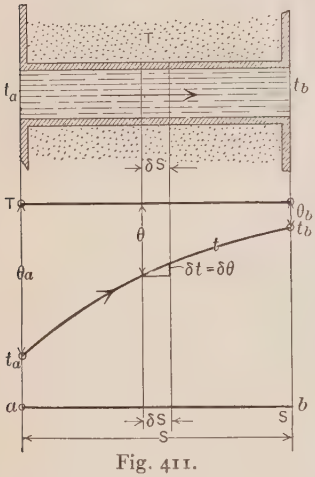


Fig. 411.

$$\delta Q = C_c W_c \delta t = C_c W_c \cdot \delta\theta;$$

but this must equal the heat transmitted *through* δS in the same time, hence

$$\delta Q = (T - t)\, K \cdot \delta S = \theta K \cdot \delta S,$$

in which T and t are average temperatures (to same scale) over δS and the other symbols have the same significance as in Sect. 305. Equating the two values of δQ in the two preceding equations gives

$$C_c W_c \cdot \delta\theta = \theta K \cdot \delta S,$$

from which

$$\int_b^a \frac{\delta\theta}{\theta} = \frac{K}{C_c W_c} \int_a^b \delta S.$$

Integration of this gives

$$\log_e \frac{\theta_a}{\theta_b} = \frac{KS}{C_c W_c}, \quad \ldots\ldots\ldots \quad (446)$$

which, rearranged and multiplied by $(\theta_a - \theta_b)$ becomes

$$C_c W_c (\theta_a - \theta_b) = \frac{KS(\theta_a - \theta_b)}{\log_e (\theta_a/\theta_b)}.$$

* These sections may be omitted in a briefer study.

Now from Fig. 411 it is seen that $(\theta_a - \theta_b) = (t_b - t_a)$ hence

$$C_c W_c (\theta_a - \theta_b) = C_c W_c (t_b - t_a);$$

but the last member of this equation is ΔQ, hence the equation above becomes

$$\Delta Q = KS \frac{(\theta_a - \theta_b)}{\log_e (\theta_a/\theta_b)} \quad . \quad . \quad . \quad . \quad . \quad (447)$$

which, in connection with Eq. (437), shows that

$$\theta_m = \frac{(\theta_a - \theta_b)}{\log_e (\theta_a/\theta_b)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (448)$$

as given in Eq. (438).

(b) For certain purposes, however, it is more convenient to write the value of θ_m in another form. From Fig. 411 it is evident that $\theta_a = (T - t_a)$ and $\theta_b = (T - t_b)$, substituting which in Eq. (448) and simplifying gives the more useful expression

$$\theta_m = \frac{t_b - t_a}{\log_e \frac{T - t_a}{T - t_b}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (449)$$

Transforming Eq. (446) gives

$$\frac{\theta_a}{\theta_b} = \log_e{}^{-1}\left(\frac{KS}{C_c W_c}\right) = e^{(KS/C_c W_c)} = n, \quad . \quad . \quad . \quad (450)$$

where n is the number whose Naperian logarithm is $KS/C_c W_c$, as given by the Log. Tables in the Appendix.

Hence for any extent of area S, the temperature difference at the end b, in terms of the known value at end a, is

$$\theta_b = \theta_a \div n; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (451)$$

and by taking different extents of area S and solving for the corresponding values of θ_b data may be obtained for plotting curves which show how the temperature difference varies as the flow progresses, for given values of K, C_c and W_c.

(c) The efficiency of heat transmission (neglecting losses) for this case was given in Eq. (441). Substituting the value of θ_b from Eq. (451) gives

$$Ef = \frac{\theta_a - \theta_b}{\theta_a} = \frac{\theta_a - \theta_a/n}{\theta_a} = 1 - \frac{1}{n}, \quad . \quad . \quad . \quad (452)$$

from which the values of the efficiencies corresponding to different extents of area S can be readily computed in any given case, and the data thus obtained can be used for constructing a curve to show the variation graphically.

308. Case II. (t = Const.) A Substance at Constant Temperature (t) Receives Heat from Another Flowing Substance whose Temperature Decreases. This is the case shown in Fig. 412 and is the reverse of Case I. The treatment is similar to the last case except that T, δT and $C_h W_h$ are substituted for t, δt and $C_c W_c$. ΔQ is given by Eq. (447) without change and θ_m by Eq. (448).

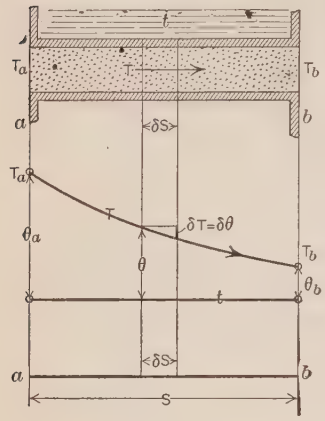


Fig. 412.

Since in this case $\theta_a = (T_a - t)$ and $\theta_b = (T_b - t)$, substitution of these quantities in Eq. (448) gives

$$\theta_m = \frac{T_a - T_b}{\log_e \dfrac{T_a - t}{T_b - t}} \quad . \quad (453)$$

Paralleling Eq. (450)

$$\theta_a/\theta_b = e^{(KS/C_h W_h)} = n, \quad (454)$$

where n is the number whose $\log_e$ is $(KS/C_h W_h)$. Using this value of n, Eqs. (451) and (452) can be applied to this case.

309. Case III. Parallel Flow in the Same Direction. (a) This case is shown by the curves in Fig. 413. As in the previous cases the heat lost by the one material in passing any area equals that transmitted through the wall and also equals that received by the second material (neglecting losses). Therefore, for an infinitesimal area, δS in the figure,

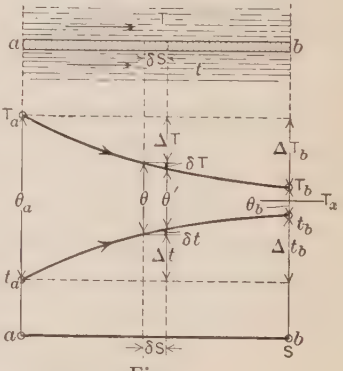


Fig. 413.

$$\delta Q = \delta T C_h W_h = \delta t C_c W_c \quad . \quad (a)$$

and for the entire area

$$\Delta Q = \Delta T_b C_h W_h = \Delta t_b C_c W_c, \quad (b)$$

where ΔT_b and Δt_b are the total charges in the temperatures of the hot and cold fluids. It is also evident from Fig. 413 that for surface δS

$$\delta T + \delta t = (\theta - \theta') = \delta\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad (c)$$

and that

$$\Delta T_b + \theta_b + \Delta t_b = \theta_a. \quad . \quad . \quad . \quad . \quad . \quad . \quad (d)$$

Equations (*a*) and (*c*) may now be used to derive two more which will be of value later. From (*a*)

$$\delta T C_h W_h - \delta t C_c W_c = 0,$$

and multiplying Eq. (*c*) by $C_c W_c$ gives

$$\delta T C_c W_c + \delta t C_c W_c = \delta\theta C_c W_c.$$

Adding these last two equations and solving gives, for the small area δS,

$$\delta T = \delta\theta \frac{C_c W_c}{C_h W_h + C_c W_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad (e)$$

which will be used later.

By analogy it is also evident that for the total area S

$$\Delta T_b = \Delta\theta \frac{C_c W_c}{C_h W_h + C_c W_c}$$

$$= (\theta_a - \theta_b)\left(\frac{C_c W_c}{C_h W_h + C_c W_c}\right). \quad . \quad . \quad . \quad . \quad (f)$$

Substituting now in Eq. (*b*) the value of ΔT_b just found gives

$$\Delta Q = (\theta_a - \theta_b) \frac{C_c W_c C_h W_h}{C_c W_c + C_c W_h}. \quad . \quad . \quad . \quad . \quad (g)$$

which will also be used later.

(**b**) Returning now to fundamentals, it is evident, as in previous cases, that

$$\delta Q = \delta T C_h W_h = K\theta \cdot \delta S.$$

From which

$$\delta T/\theta = (K \cdot \delta S)/(C_h W_h).$$

Substituting for δT its value from Eq. (*e*) and rearranging gives

$$\frac{\delta\theta}{\theta} = K \cdot \delta S\left(\frac{C_h W_h + C_c W_c}{C_h W_h C_c W_c}\right)$$

and integrating between the limits *a* and *b* yields

$$\log_e \frac{\theta_a}{\theta_b} = KS \frac{C_h W_h + C_c W_c}{C_h W_h C_c W_c}. \quad . \quad . \quad . \quad . \quad . \quad (h)$$

Multiplying both sides by $(\theta_a - \theta_b)$ and rearranging gives

$$(\theta_a - \theta_b) \frac{C_c W_c C_h W_h}{C_h W_h + C_c W_c} = KS \frac{(\theta_a - \theta_b)}{\log_e (\theta_a/\theta_b)}. \quad . \quad . \quad (i)$$

But from Eq. (*g*) it is seen that the left-hand member is ΔQ, hence, as in the other cases,

$$\Delta Q = KS \frac{(\theta_a - \theta_b)}{\log_e (\theta_a/\theta_b)}. \quad . \quad . \quad . \quad . \quad . \quad (455)$$

Comparison with Eq. (437) shows that

$$\theta_m = \frac{(\theta_a - \theta_b)}{\log_e (\theta_a/\theta_b)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (456)$$

as in the other cases.

From Eq. (*h*)

$$\theta_b = \theta_a \div e^{KS\left(\frac{C_c W_c + C_h W_h}{C_c W_c C_h W_h}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (457)$$

$$= \theta_a \div n, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (458)$$

where n is the number whose $\log_e$ is $KS\left(\frac{C_c W_c + C_h W_h}{C_c W_c C_h W_h}\right)$.

This last equation makes it possible to determine θ_b when θ_a and S are known.

(c) From Fig. 413 it is apparent that $\Delta T_b = (T_a - T_b)$. Substituting this in equation (*f*), putting $\theta_b = \theta_a \div n$, and solving gives

$$T_b = T_a - \theta_a\left(1 - \frac{1}{n}\right)\left(\frac{C_c W_c}{C_c W_c + C_h W_h}\right). \quad . \quad . \quad (459)$$

And by analogy

$$t_b = t_a + \theta_a\left(1 - \frac{1}{n}\right)\left(\frac{C_h W_h}{C_c W_c + C_h W_h}\right). \quad . \quad . \quad (460)$$

Then if $\theta_b = 0$, which occurs when $S = \infty$, T_b becomes equal to t_b and equal to the limiting temperature T_x. Thus

$$T_x = T_a - \theta_a\left(\frac{C_c W_c}{C_c W_c + C_h W_h}\right), \quad . \quad . \quad . \quad . \quad (461)$$

or

$$T_x = t_a + \theta_a\left(\frac{C_h W_h}{C_c W_c + C_h W_h}\right). \quad . \quad . \quad . \quad (461a)$$

Since the maximum amount of heat that can be transmitted is $(T_a - T_x)\ C_h W_h = (T_x - t_a)\ C_c W_c$, the *true* efficiency is therefore, in the ideal case,

$$TEf_h = \frac{(T_a - T_b)\ C_h W_h}{(T_a - T_x)\ C_h W_h} = \frac{(T_a - T_b)}{(T_a - T_x)}, \quad . \quad . \quad . \quad (462)$$

or

$$TEf_c = \frac{(t_b - t_a)\ C_c W_c}{(T_x - t_a)\ C_c W_c} = \frac{t_b - t_a}{T_x - t_a}. \quad . \quad . \quad . \quad . \quad (463)$$

As given in Eqs. (443) and (442) the comparative efficiency (for comparison with the other cases) is

$$CEf_h = \frac{T_a - T_b}{\theta_a}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (464)$$

or

$$CEf_c = \frac{t_b - t_a}{\theta_a}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (465)$$

depending on whether the object is to cool the hot fluid or to heat the cold one.

310. Case IV. Counter Flow. (a) This case is shown in Fig. 414, the directions in which the temperature curves are generated being shown by the arrows. Compared with the other cases it is a little more difficult to develop usable equations for this sort of flow because, in general, only the initial temperatures T_a and t_b at opposite ends of the plate are known and both θ_a and θ_b are unknown.

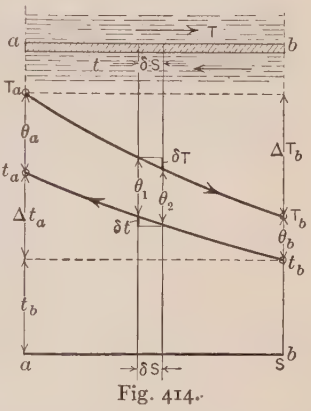


Fig. 414.

(b) As in previous cases, however,

$$\delta Q = \delta T C_h W_h = K\theta \cdot \delta S \quad . \quad (a)$$

$$= \delta t C_c W_c . \quad . \quad . \quad . \quad . \quad . \quad (b)$$

Hence

$$\delta t/\delta T = \frac{C_h W_h}{C_c W_c} = Z, \quad . \quad . \quad (c)$$

the symbol Z being introduced as this ratio will be frequently used in the following development. From the figure it is seen that the change in temperature difference over any elementary area δS is

$$\delta\theta = \theta_1 - \theta_2 = \delta T - \delta t.*$$

Substituting for δt in terms of δT from Eq. (c) and solving gives

$$\delta T = \delta\theta/(1 - Z) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (d)$$

If this is substituted in Eq. (a) there results, after transformation,

$$\frac{\delta\theta}{\theta} = \frac{1 - Z}{C_h W_h} K \cdot \delta S,$$

which, being integrated between limits a and b, gives

$$\log_e \frac{\theta_a}{\theta_b} = \frac{1 - Z}{C_h W_h} KS, \quad . \quad . \quad . \quad . \quad . \quad (466)$$

which, after both sides are multiplied by $(\theta_a - \theta_b)$ and rearranged, gives

$$(\theta_a - \theta_b)\frac{C_h W_h}{1 - Z} = \frac{KS(\theta_a - \theta_b)}{\log_e(\theta_a/\theta_b)} . \quad . \quad . \quad . \quad (467)$$

(c) From the figure it is further apparent that

$$\Delta T_b + \theta_b = \Delta t_a + \theta_a;$$

and from Eq. (c), by analogy $\Delta t_a = Z\Delta T_b$, ... (e)

* The analysis given is for $Z < 1$. If $Z > 1$, then in Fig. 414 $\theta_b > \theta_a$ and $\theta_2 > \theta_1$. For this case substitute $(\theta_2 - \theta_1)$ for $(\theta_1 - \theta_2)$, $(\delta t - \delta T)$ for $(\delta T - \delta t)$, $(Z - 1)$ for $(1 - Z)$, θ_b/θ_a for θ_a/θ_b, and $(\theta_b - \theta_a)$ for $(\theta_a - \theta_b)$.

substituting from which in the last equation and solving gives

$$\Delta T_b = (\theta_a - \theta_b) \div (1 - Z). \quad . \quad . \quad . \quad . \quad . \quad (f)$$

Now $\Delta Q = \Delta T_b C_h W_h$ and substituting the value of ΔT_b just found gives

$$\Delta Q = (\theta_a - \theta_b) \frac{C_h W_h}{1 - Z}. \quad . \quad . \quad . \quad . \quad . \quad (g)$$

(d) Returning now to Eq. (467) and comparing with Eq. (g), it is obvious that the left-hand member is equal to ΔQ, hence

$$\Delta Q = \frac{KS\,(\theta_a - \theta_b)}{\log_e (\theta_a/\theta_b)}. \quad . \quad . \quad . \quad . \quad . \quad (468)$$

and

$$\theta_m = \frac{\theta_a - \theta_b}{\log_e (\theta_a/\theta_b)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (469)$$

These expressions are evidently the same as in the other cases. However they are not of value until θ_a and θ_b have been determined, either by the methods which will now be given, or from actual experiment with existing apparatus.

(e) From Eq. (466) it is evident that

$$\theta_a/\theta_b = e^{\frac{1 - Z}{C_h W_h} KS} = n, \quad . \quad . \quad . \quad . \quad . \quad . \quad (470)$$

in which n is the number whose Nap. log is $\frac{1 - Z}{C_h W_h} KS$. Substituting the value of θ_b from this equation in Eq. (f) gives *

$$\Delta T_b = \theta_a \left(1 - \frac{1}{n}\right) \div (1 - Z). \quad . \quad . \quad . \quad . \quad (h)$$

From Fig. 414 it can be seen that $\Delta t_a = (T_a - t_b - \theta_a)$, substituting which in Eq. (e) and solving gives

$$\Delta T_b = (T_a - t_b - \theta_a)/Z. \quad . \quad . \quad . \quad . \quad . \quad . \quad (i)$$

Now subtracting Eq. (i) from Eq. (h) and solving gives

$$\theta_a = \frac{T_a - t_b}{Z} \div \left(\frac{1}{Z} + \frac{1 - \frac{1}{n}}{1 - Z}\right), \quad . \quad . \quad . \quad . \quad (471)$$

by which the temperature difference at one end can be determined in terms of known quantities.

* For $Z > 1$ the footnote on p. 647 applies up to this point. Change (h) to be $\Delta T_b = \theta_a\,(n - 1) \div (Z - 1)$ and in (471) substitute $\left(\frac{n - 1}{Z - 1} + \frac{1}{Z}\right)$ in the parenthesis.

(f) With θ_a known, the value of θ_b follows from Eq. (470); ΔQ and θ_m can be determined from Eqs. (468) and (469); and the final temperatures of cold and hot fluids are

$$t_a = (T_a - \theta_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (472)$$

and

$$T_b = (t_b + \theta_b) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (473)$$

(g) It is important to note that the expressions in (f) can be used only to determine the conditions at the *ends a* and *b* or over

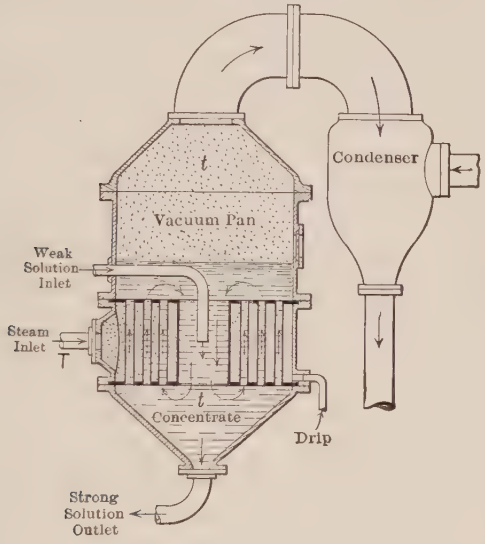


Fig. 415.

the entire length of area S. They cannot be used for intermediate points, that is, for plotting the curves T_aT_b or t_at_b and the like.

311. **Case V. (T = const. & t = const.) A Hot Substance Surrenders Heat at Constant Temperature to a Cold Substance whose Temperature is Constant.** (a) This case is exemplified in vacuum pans in which steam at constant temperature (T) surrenders some of its latent heat to evaporate a solution at constant temperature (t). The type of apparatus which is known as a "single effect vacuum pan" is illustrated in Fig. 415 in one of its many forms and arrangements.

(b) In such case $\theta_m = (T - t)$ and the heat transmitted through the heating surfaces is $\Delta Q = KS\theta_m$ from Eq. (437).

Neglecting losses, the weight of steam condensed in unit time by an amount of heat equal to ΔQ is obviously

$$W_h = \Delta Q/r, \quad . \quad . \quad . \quad . \quad . \quad . \quad (474)$$

where r is the latent heat of steam at temperature T °F.; and the weight of solution evaporated in unit time is

$$W_e = \Delta Q/[\lambda_e - (t_0 - 32)], \quad . \quad . \quad . \quad . \quad (475)$$

where λ_e is the total heat (above 32°) per pound of the vapor formed at temperature t from the solution, and t_0 is the temperature at which the solution enters the vacuum pan.

(c) The vapor (at temperature t_1) from the solution in one vacuum pan may be used, as in Fig. 416, to vaporize (at lower

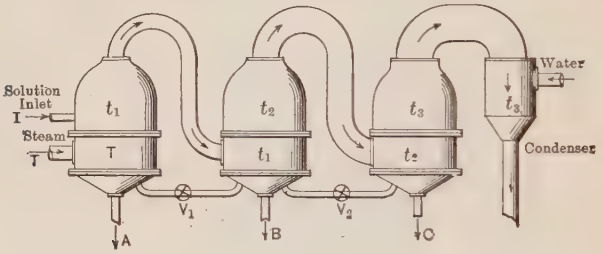


Fig. 416.

temperature t_2) the solution in a second vacuum pan, the latter acting as condenser for the first element; the vapor from the second pan (at temperature t_2) may be similarly used to evaporate (at lower temperature t_3) the solution in a third pan, and so on, the vapor from the last pan being carried to a condenser. When more than one pan is thus used the arrangement is termed "multiple effect." Arrangements are called "double effect," "triple effect," "quadruple effect," and so on according to the number of pans in the series. In Fig. 416 the weak solution is admitted at I to the first pan from which it is fed at proper rate through valves V_1 and V_2 to the other ones. The strong solution is withdrawn from the respective pans at A, B and C.

CHAPTER XXXVI.

APPARATUS FOR HEATING FEED WATER.

312. Object of Heating Feed Water. (a) The principal advantages to be derived from heating the feed water supplied a boiler are:

(1) A decrease in the amount of fuel required to generate the steam, hence an increase in the over-all efficiency of the plant; (2) less severe strains in the boiler metal, as there is less difference of temperature between boiler shell and the fresh feed water; (3) the partial deposition outside of the boiler of scale-forming impurities contained in the water; and (4) an increase in the steaming capacity of the boilers as less heat need be transmitted per pound of steam generated.

(b) An idea of the saving of fuel derived from the use of hot boiler feed can be obtained by analyzing an average case. Assume, for instance, that the water as received at the plant has an average temperature of 60° F. with $q = 28.08$ and that it is converted into steam at a pressure of 150 lbs. abs. (with $\lambda = 1193.4$), thus requiring the addition of $1193.4 - 28.08 = 1164$ B.t.u. (approx.) per pound. Assuming the specific heat of water as unity, every 11.64 (say 12) degrees by which the temperature of the water is raised before its introduction into the boiler means 1 per cent less heat to be added in the boiler, which would roughly correspond to a saving of 1 per cent of fuel.

Expressed as a formula, the theoretical saving, due to using hot feed water at temperature t_f' instead of cold at temperature t_f, in a boiler generating *dry saturated* steam is approximately:

$$\text{per cent saving} = \frac{q_f' - q_f}{\lambda - q_f} \times 100, \quad . \quad . \quad . \quad (476)$$

in which

q_f' = sensible heat of the hot feed water above 32° F. $= (t_f' - 32)$ approx.,

q_f = sensible heat of the cold feed water above 32° F. $= (t_f - 32)$ approx.,

λ = total heat of steam above 32° F. at boiler pressure.

If the steam is *superheated* in the boiler, the saving is

$$\text{per cent saving} = \frac{q_f' - q_f}{C_p D + \lambda - q_f} \times 100. \quad . \quad . \quad (477)$$

The savings in per cent resulting from different amounts of feed-water heating with different initial temperatures for the case of saturated steam are shown diagrammatically in Fig. 417. These are obtained by the method just given.

(c) Inspection of these curves will show that if water at as low a temperature as 40° F. is raised to a temperature of 200° F.,

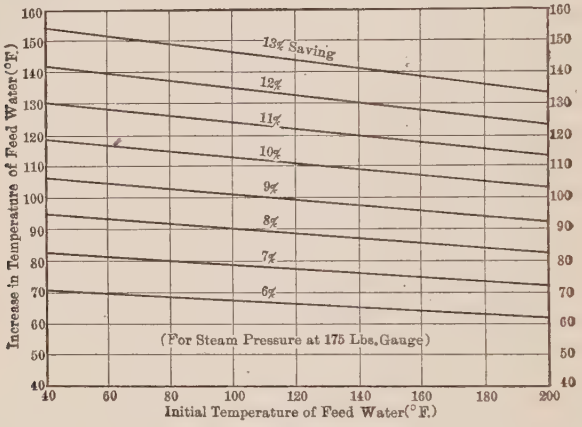


Fig. 417.

i.e., through a range of 160° F., the saving will be slightly over 13 per cent; and a change from 60° to 180° effects a saving of about 10 per cent of the fuel that would otherwise be needed.

As the boilers do not have to transmit as much heat per pound of steam generated from preheated feed water as from cold water, smaller or fewer boilers may be used for a given output, when other considerations permit.

313. Feed-Water Heaters in General. (a) One method of heating the boiler feed water is by using for that purpose some of the latent heat in the exhaust steam from a steam-driven prime mover (as has already been explained in Section 297), the apparatus in which the transmission occurs being called a *Feed-Water Heater*, or an Exhaust Steam Feed-Water Heater.

(b) If w pounds of steam are utilized for heating per pound of *raw* feed water which is at temperature t_f, then

$$(t_f' - t_f) = w\,[\lambda - (t_f' - 32)] \times Ef,$$

in which t_f' = temperature finally attained by the feed water and condensed steam,

λ = total heat above 32° of 1 lb. of steam used for heating, and

Ef = efficiency of heater.

Then the temperature of the feed water, when w lbs. of steam are used per pound of feed, is

$$t_f' = \frac{t_f + w\,(\lambda + 32)\,Ef}{1 + wEf}, \quad . \quad . \quad . \quad . \quad (478)$$

and to attain a temperature of t_f' the weight of steam required per pound of *raw* feed is

$$w = \frac{t_f' - t_f}{[\lambda - (t_f' - 32)]} \div Ef. \quad . \quad . \quad . \quad . \quad (479)$$

In the foregoing it has been assumed that the condensed steam is finally cooled in the heater to the temperature of the outgoing feed water. Any error which is thus introduced is corrected by the efficiency factor.

(c) *If the condensed steam is not returned to the boiler* with the feed, this expression for w also gives the maximum proportion of the total steam generated which can be utilized for heating an equivalent weight of feed water. For example, if feed water at 60° is heated to 212° F. by steam from an engine exhausting at atmospheric pressure, and if $Ef = .90$, the maximum possible weight of steam so utilized is found to be about 17 per cent, or about $\frac{1}{6}$th of all that is available. It is, of course, desirable that the heat in the remaining portion ($\frac{5}{6}$) should be utilized in some other way as far as is possible.

In plants in which the main units are operated condensing, the auxiliary engines, which generally use less than $\frac{1}{6}$ of the total steam, are operated noncondensing and their exhaust steam is utilized for feed heating. This results in greater thermal efficiency of the plant as a whole than would exist if the auxiliaries were connected to the condenser, since all the heat of the steam not used for power is then theoretically returned to the boiler.

(d) *When the condensed steam* (at t_f') *is returned to the boiler* with the feed, the total weight of feed per pound of raw feed is $(1 + w)$, and the proportion of raw to total is $1/(1 + w)$, where w is the weight of steam condensed per pound of *raw* feed, as given by Eq. (479). Then the steam condensed per pound of *total* feed water is

$$w' = w/(1 + w) \quad . \quad . \quad . \quad . \quad . \quad . \quad (480)$$

which is the maximum proportion of the *total* steam generated that can be utilized for feed heating in such cases.

(e) If the heater is located in the exhaust system between the prime mover and the condenser, it is called a (1) *Vacuum Heater*, and the maximum temperature which the feed water can theoretically attain is that corresponding to the vacuum. When located in the exhaust system of a noncondensing unit, it is termed an (2) *Atmospheric Heater*, and the theoretical temperature attainable with sufficient steam is 212°. Should the heater take steam from the auxiliary apparatus of the power plant it may be named an (3) *Auxiliary Heater*. If the condensate from the main units is not used as feed and there is not enough steam from the auxiliary apparatus to raise the temperature to the maximum otherwise attainable, the vacuum and auxiliary heaters are sometimes arranged in series — the water first passing through the former, which is then called a (4) *Primary Heater*, and finally through the latter, which becomes the (5) *Secondary Heater*. When the pressure of the steam used for heating is considerably above atmospheric, the apparatus is a (6) *Pressure Heater*.

When the arrangement is such that the feed water and steam intermingle in the same chamber the heater is said to be of the (7) *Open Type;* and when the two substances are kept separate by heat-transmitting surfaces it is of the (8) *Closed Type*.

314. Open Heaters. (a) Heaters of this type are generally in the form of rectangular boxes, or circular shells, fitted with coarse cascading or spraying devices to break up the water as it passes through and thus bring it into more intimate contact with the steam. They usually contain a filtering bed or settling chamber in which the solids carried in suspension or in solution are more or less completely removed after heating. When necessary, they are also fitted with oil separators in the steam

inlet for removing the cylinder oil from the steam before it comes in contact with the water. This oil, if carried over to the boiler, would seriously reduce the transmission of heat in that apparatus

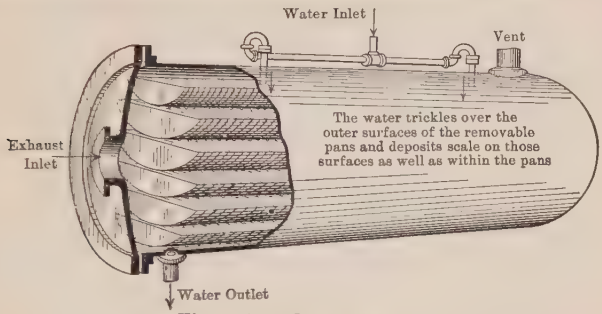

Water Inlet
Vent
Exhaust Inlet
The water trickles over the outer surfaces of the removable pans and deposits scale on those surfaces as well as within the pans
Water Outlet

Fig. 418. — Open Heater.

and might even cause overheating of the metal parts subjected to high temperatures. Two of the great variety of heaters of this type are shown in Figs. 418 and 419, the water level in the latter being automatically regulated by a float.

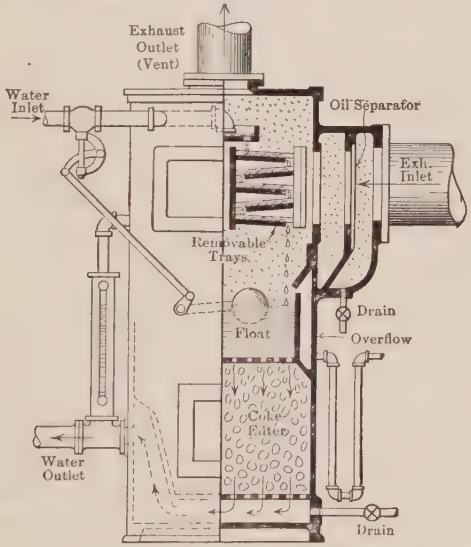


Fig. 419. — Open Heater.

(b) The main advantages of Open Atmospheric Heaters are: (1) the feed water can be heated nearly to 212° F. if sufficient steam is available, and surplus steam can be utilized for heating

buildings and in industrial processes where conditions permit; (2) scale and oil do not affect the surrender of the heat; and (3) the hot condensed steam is returned to the boiler with the raw feed, but should be purified of oil (if any is brought over from the engine cylinder by the steam).

The open heater may be arranged, as in Fig. 419, to include in its structure (*a*) an oil separator, which is usually located in the exhaust pipe at entrance to heater, (*b*) a filter for removing sediment, part of which may be precipitate brought down by heating, and (*c*) a hot well, which may also receive the returns (condensate) from systems for heating buildings and such. As the feed water is hot and at pressure near atmospheric these heaters should be located above the feed pump and this latter

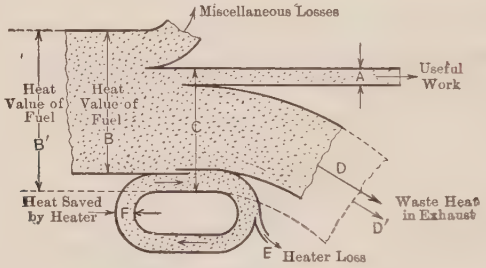


Fig. 420.

should be suitable for pumping hot water. If the raw water is not available under sufficient head to flow into the heater, a second or cold-water pump must be added.

(**c**) The proportion of the total steam generated that can be used for heating the feed water is obtainable from Eq. (479), being about $\frac{1}{6}$th in ordinary cases, and the saving of fuel effected is given by Eq. (476) or (477), the maximum being about $\frac{1}{8}$th. Fig. 420 shows in full lines the energy stream for a power plant having all units exhausting to atmosphere but using as much waste heat as is possible in a feed-water heater; and in dotted lines it illustrates the same case when no heater is used. In the first case the over-all thermal efficiency is A/B and in the second A/B', the heat supplied to the prime mover being C in both cases.

(**d**) If the main engines are condensing, and the steam from steam-driven auxiliary apparatus is used for feed heating, the

stream line is that illustrated in Fig. 421 by the heavy lines. A is the heat utilized, B is the heat value of the fuel used, C is the heat supplied to the main and auxiliary engines, and the ratio A/B is the over-all thermal efficiency.

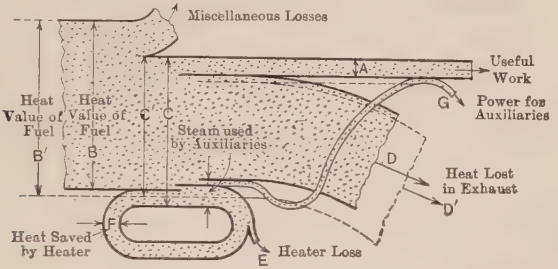


Fig. 421.

But if the auxiliaries are power driven, the energy being furnished by the main units, then the case is shown in the same figure by the dotted lines; the useful output is A (as before), B' is the heat value of the fuel used, and C' is the heat furnished to the main units, its amount being less than C because the water rates of the larger units are lower than those for the small auxiliary engines. The over-all thermal efficiency in this case is A/B' which is less than when the auxiliaries are driven by steam and their exhaust is used for feed heating.

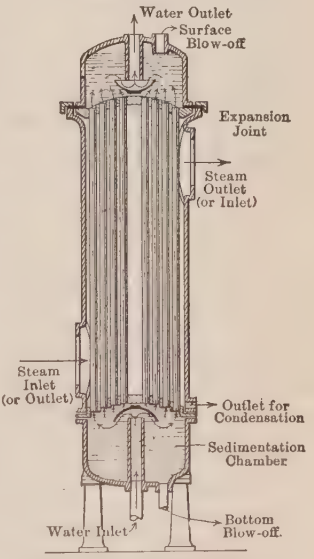


Fig. 422. — Closed Heater.

315. Closed Heaters. (a) Heaters of this type are so arranged that the steam does not come in contact with the water. They are generally constructed with straight or coiled tubes contained in a shell of some sort, with proper provision being made for inequalities in expansion. The water generally passes through the tubes or coils and the steam fills the enveloping space, the condensation being drained off as it collects. The three most common of the many possible arrangements are shown in Figs. 422 to 424.

(b) Closed heaters are comparatively difficult to clean as a large part of the impurities in the water is deposited on the inside of the tubes and forms a coating similar to boiler scale. To counteract this effect, the water is often forced through the tubes at enormously high velocity, tending to keep them clean by "scouring." However the power required for pumping places a practical limit to the velocities used and the method is only partly successful. If the steam contains oil, and it is not removed before entering the heater, the tubes will become coated

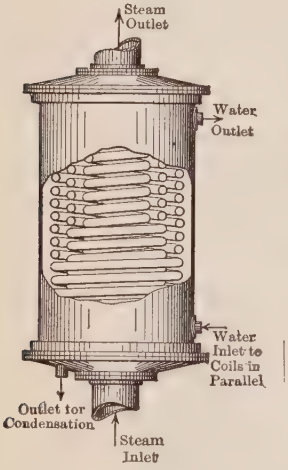


Fig. 423. Coil Heater.

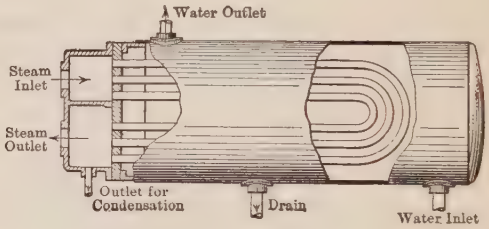


Fig. 424. — Closed Heater.

with this material, which is of low conductivity, and the rate of heat transmission will be greatly impaired.

(c) These heaters are generally placed between the feed pump and the boiler, hence the former deals with cold water only and but one pump is required (not considering reserve pumps for emergencies) as against two with open heaters which do not receive the raw water under head. The feed water is free from oil, but the condensate is generally wasted. Figs. 420 and 421 apply equally well to this case except that the heater loss E includes the sensible heat of the condensate (measured above the temperature of the raw feed) unless there is more than enough steam to raise the feed temperature to the maximum possible value. The maximum temperature attainable is gen-

erally 2 degrees or more below the steam temperature, which latter depends on the pressure of the steam used.

(**d**) In all such cases, the auxiliary apparatus, such as heaters, should be so piped that when out of commission the steam can be exhausted to atmosphere direct, and the feed water can be by-passed around the heater. The piping is therefore arranged somewhat as in Fig. 425, *S* and *W* being the steam and water by-pass valves which are normally closed.

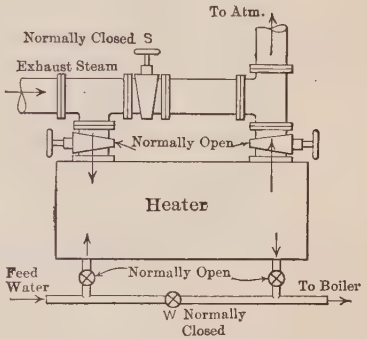


Fig. 425.

(**e**) The saving effected by closed heaters can be found from Eqs. (476) and (477), the temperature attained by the feed water is given by Eq. (478), and the proportion of the total steam generated, that can be used for heating the raw feed, is given by Eq. (479).

(**f**) The heat transmission falls under Case I of Sections (306) and (307). The mean temperature head is given by Eqs. (438) or (449) and the number of square feet of heating surface can be found from Eq. (440) when K is known. For feed-water heaters K varies widely with the kind of material, character of surface (scale, oil film, corrugations, etc.), with the velocity of flow of feed water and with other factors. It ranges ordinarily with copper, or brass, tubes from 175 with velocity of $12\frac{1}{2}$ feet per minute and single pass, to 250 with velocity of 50 ft. per min. and multipass; with a velocity of 150 ft. per min. through coils it reaches 300 B.t.u. per square foot per hour per degree difference in temperature, while under very favorable conditions much higher values are attainable.

(**g**) Closed heaters with copper tubes are sometimes rated in terms of "*heater horse power*," $\frac{1}{3}$ square foot of surface being allowed per rated horse power. On this basis, if the steam pressure is atmospheric and if $K = 192$, the 30 lbs. (approx.) of feed water required per so-called boiler horse power will be heated from 60° to 194° (134° increase) by the $\frac{1}{3}$ square foot of surface — or 1 sq. ft. will heat about 90 lbs. of water (the amount required for 3 boiler h.p.) through this temperature range.

316. Economizers. (a) The function of the economizer is to abstract a portion of the heat from the flue gases, and to deliver it to the feed water on its way to the boiler, thereby somewhat reducing the large loss *c* in Fig. 3. The energy stream for this

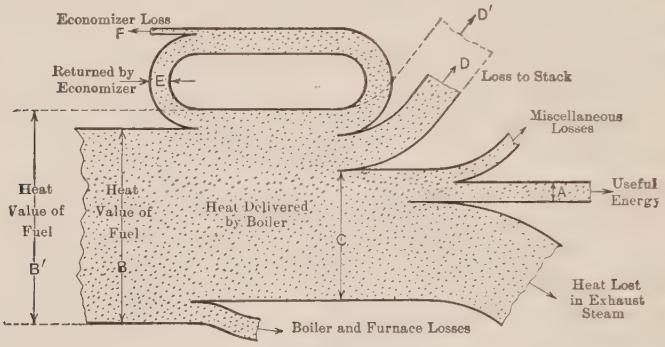


Fig. 426.

case is shown by the full lines in Fig. 426, in which the case without the economizer is that with dotted lines, the boiler performance being assumed to remain unchanged. For the same boiler-output the fuel used is in the ratio of B to B'.

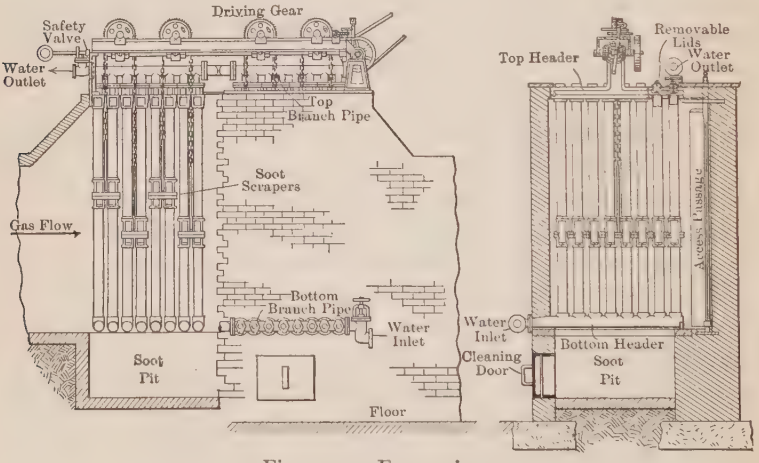


Fig. 427. — Economizer.

(b) One form of economizer is shown in Fig. 427. The apparatus usually consists of nests of staggered, vertical, cast-iron tubes fitted into top and bottom headers (with metal to metal

joints), each set of headers being connected together by longitudinal branch pipes having handholes which give access to the interior for washing out deposits. In the upper headers are located removable lids opposite the ends of the tubes in order to give access to the latter, and power-driven scrapers constantly move along the external surfaces of the tubes to remove the deposit of soot, the scrapings falling to a pit below, from which they are withdrawn from time to time. The water is preferably introduced at the end farthest from the boiler and discharged from the nearer end; for its direction of flow is then counter to that of the flue gases, thus obtaining the counterflow of Case IV, discussed in Chapter XXXV. The setting is either of brick or of sheet steel lined with nonconducting material (magnesia or asbestos); the arrangement of flues is such (see Fig. 428) that the gases from the boilers can be by-passed direct to the stack when the economizer is out of commission, and the water can be delivered direct to the boiler.

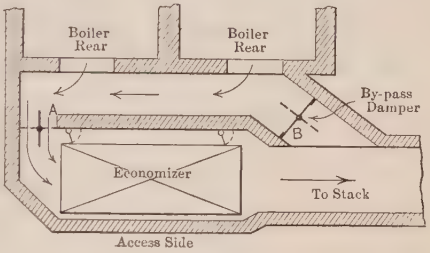


Fig. 428.

(**c**) In some instances the flue gases are cooled from ordinary stack temperature of 550° to 650° F. to as low as 240° F. and the water is heated to 270° or more; temperatures much higher than can be obtained with an atmospheric feed-water heater. But because the temperature of the stack gases is low and because of the additional resistance in the flues due to the presence of the economizer tubes, the natural draft must generally be assisted in some manner. Hence, in connection with the financial problem involved, the cost of such draft apparatus and the annual expenses chargeable against it must be added to the charges against the economizer itself, including those for the space it occupies and the power required for driving the scrapers. As the economizer occupies a great deal of space it is frequently placed either above the boiler or outside of the building.

(**d**) In addition to the four advantages accruing in all cases from heating feed water, as given in Section 312 (**a**), the econo-

mizer has (5) a great reserve of hot water near the vaporizing point, ready to meet sudden demands on the boiler; (6) its use may make it possible for the boiler itself to operate with higher efficiency, and (7) it is especially advantageous when the boilers are being forced, for then the flue gases are hottest and the stack waste is ordinarily the greatest. Owing to the higher temperature attained by the water, some scale-forming materials are deposited which are not precipitated in atmospheric feed heaters.

(e) If the counterflow principle is used in the economizer the equations of Section 310 apply. A simple approximation can be made however by assuming the two curves in Fig. 414 to be straight lines, then, at the middle of the curves,

$$\theta_m = (T_a - \Delta T_b/2) - (t_b + \Delta t_a/2) = T_a - t_b - \tfrac{1}{2}(\Delta T_b + \Delta t_a). \quad \text{(a)}$$

But
$$C_p G \Delta T_b = w \Delta t_a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(b)}$$

where C_p = specific heat of gas (= 0.24),

G = weight of gas per boiler h.p.-hr.,

w = pounds of water per boiler h.p.-hr.

Solving (b) for ΔT_b and substituting in (a) gives

$$\theta_m = T_a - t_b - \tfrac{1}{2}\Delta t_a (1 + w/GC_p). \quad . \quad . \quad . \quad \text{(c)}$$

But the heat absorbed in the economizer by the water used per boiler horse power is $Q = w\Delta t_a$, hence Eq. (437) becomes (per boiler h.p.)

$$w\Delta t_a = KS\theta_m. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(d)}$$

Substituting the value of θ_m from (c) and solving gives the increase in the temperature of feed water as

$$\Delta t_a = \frac{S(T_a - t_b)}{\dfrac{w}{K} + \left(\dfrac{w + GC_p}{2GC_p}\right)S} \text{ (approx.)}. \quad . \quad \text{(481)}$$

In practice S ranges from $2\frac{1}{4}$ to 5 square feet per boiler horse power. Corresponding to gas temperatures of 300° and 600° F. respectively K has values of about $2\frac{1}{4}$ and $3\frac{1}{4}$ B.t.u. per square foot per degree difference of temperature per hour. The weight of water (w) per boiler-horse-power hour is generally taken at

about 30 lbs.; and the weight of gas (G) per boiler-h.p./hr. as $G = (1 + 11.6\,x) \times F$,

where x = excess coefficient,

F = weight of combustible per boiler-h.p./hr. = 3 to 4 lbs.

G is ordinarily from 80 to 120 lbs. per boiler h.p./hr. Then the final temperature of the feed water is

$$t_f = t_b + \Delta t_a, \quad . \quad . \quad . \quad . \quad . \quad (482)$$

and the final temperature of the flue gas is

$$T_b = T_a - \Delta T_b, \quad . \quad . \quad . \quad . \quad . \quad (483)$$

in which

$$\Delta T_b = \frac{w \Delta t_a}{G C_p}; \quad . \quad . \quad . \quad . \quad . \quad (484)$$

and if $w = 30$, $G = 80$, and $C_p = 0.24$

$$\Delta T_b = 1.56\,\Delta t_a. \quad . \quad . \quad . \quad . \quad . \quad (485)$$

(f) The per cent saving effected by raising the feed temperature by the amount Δt_a may be obtained from Eqs. (476) and (477) by substituting Δt_a for the numerator. The actual saving of boiler and economizer taken together may be still more, since the boiler may have higher efficiency because of the better conditions of operation.

NOTE: See "Economizer Equation" and curves, by A. W. Smith in Sibley Journal, Jan., 1916, and the later discussion in the June issue.

CHAPTER XXXVII.

CONDENSERS AND RELATED APPARATUS.

317. Advisability of Condensing. The principal advantages accruing from the use of condensers in connection with steam-driven prime movers are: (1) Improved thermal efficiency of the unit (except in the smaller sizes); (2) greater power from a given size of prime mover; and, when the condensate is used for feed water, there are the additional advantages of (3) the thermal gain from using hot feed water and (4) the freedom from deposits of scale in the boiler because the feed water is distilled.

However, despite the apparent advantages, it is not always desirable to operate condensing, for financial and other reasons. The additional expense for the extra equipment, its installation, attention and upkeep, the expenditure for condensing water, for pumps and their operation, and the additional space required by the apparatus may in some cases wholly offset the advantages. It is generally not considered profitable to operate condensing with small engines, or with simple engines of the ordinary types (some special types, such as the unidirectional flow engine, operate to best advantage when condensing): nor should condensers ordinarily be used when a considerable part of the exhaust steam can be employed for heating or for industrial purposes.

318. Condensers in General. (a) The two main classes into which all types of condensers may be divided are: (1) *Direct-contact Condensers* and (2) *Surface Condensers*. In the former, the steam and condensing water mingle in the same chamber, while in the latter type they are kept separated by heat-transmitting surfaces. In each class there are many different arrangements possible and some of these will be considered in detail later.

(b) Theoretically, the material handled by a condenser is low-pressure steam; actually it is a mixture of water, water vapor

and air. Part of this air comes from the boiler, being carried into that vessel in solution in the feed water, and part of it leaks into the system through the stuffing boxes surrounding the piston and valve rods, through the joints of pipes and of such other parts of the equipment as are handling the material below atmospheric pressure. Also the water used for condensing carries air in solution when under atmospheric conditions, and in direct-contact condensers this air is released under diminished pressure and is added to that which enters in the various ways just outlined.

(c) Then, according to Dalton's law, the total pressure within the condenser is the combination of the pressures of the air and vapor, *i. e.*, it is the sum of their partial pressures. The importance of this fact is best appreciated from an example.

Assume the temperature within a condenser to be 115° F. Then if the condenser contained only water and saturated steam at this temperature the pressure within the enclosure would be 2.99 inches of mercury, corresponding to a vacuum of 26.93 inches. If, however, every pound of steam has mixed with it one-quarter of a pound of air, which is not at all uncommon, the pressure due to this air can be found as follows:

One pound of saturated steam at a pressure of 2.99″ Hg. occupies a volume of 231.9 cu. ft. This must also be the volume occupied by the 0.25 lbs. of air mixed with it, and the temperature of this air is that of the steam (115°). Then from the law of ideal gases, the pressure of the air in the condenser is

$$P = \frac{RT}{V} = \frac{53.34 \times (115 + 460)}{4 \times 231.9} = 33.06 \text{ lbs./sq. ft.},$$

or $$p = 0.46 \text{ inches of mercury.}$$

Thus, the total pressure in the condenser will then be 2.99″ + 0.46″ = 3.45 in. Hg. and the vacuum will be 29.92 − 3.45 = 26.47 in. Hg. The back pressure on the prime mover is slightly higher than this, as a pressure drop must exist between that piece of apparatus and the condenser in order to cause the steam to flow through the exhaust pipe.

With one pound of air per pound of steam, which is a possible condition, the pressure due to the air would be 1.84 inches of mercury, under the same circumstances, and the vacuum would only be 25.09 inches.

The air is thus seen to have a very appreciable effect upon the

vacuum and every precaution should therefore be taken to prevent an excessive amount of it entering the apparatus. If allowed to accumulate it would gradually increase in pressure and destroy the vacuum. It must therefore be removed as rapidly as it collects. Before it can be discharged from the condenser, however, its pressure must be raised to that of the atmosphere (or slightly above) which is done by an air compressor or pump having terminal pressures sufficiently above atmospheric to effect a discharge. When this pump handles only the air it is called an "*Air Pump*," or "*Vacuum Pump*."

In some cases where the water is discharged from the condenser with considerable velocity, the arrangement is such that the air is ejected by the water, no separate air pump being needed.

(**d**) Condensers in steam-power plants practically always use water as the condensing medium, but any liquid or gas that could be obtained cheaply, in sufficient quantities and at a low temperature, could be used; in fact air has been so utilized in a number of special instances.

It is seldom that water is available under a head sufficient to cause it to flow into or through a condensing apparatus. It is therefore generally delivered to the condenser by a "*Circulating Pump*," which may be independently driven by steam, by electric motor or by belt, or may be operated by links driven by the prime mover. These pumps generally have comparatively low lifts and handle large volumes, hence the centrifugal type is commonly used, although there are many cases where the rotary or the reciprocating types have the advantage and are installed.

In apparatus in which condensing water and steam mix and form a vacuum, the condensing water is often forced into the condenser by the atmospheric pressure acting on the surface of the water outside, no circulating pump being used. This is very common practice where the suction head is not over 15 feet, and it is used even with greater heads in some instances.

(**e**) The removal of the water from the vacuum chamber of the condenser may be accomplished in several ways. If the hot well, which receives the condensate, can be located with water level at least 34 feet below the condenser, the water can be discharged by gravity through a "Tail Pipe," or "Barometric Tube," whose lower end is submerged in the hot well (the 34-foot column of water corresponding to a 30 inch column of Hg.

on the barometer). In other cases it is necessary to have pumps which raise the water from condenser pressure to atmospheric. Such pumps are called "*Tail Pumps*," "*Hot-well pumps*," etc., when they handle only water (and whatever air it happens to have entrapped). In many instances, however, the same pump plunger discharges both the water and the free air, in which case the one pump serves both as hot-well pump and as air pump, and is then called a "*Wet Vacuum Pump*" or "*Wet Air Pump*."

(f) Each prime mover may have its *independent condenser* or there may be a *central condensing equipment* for a number of units. In the former case the exhaust piping may be made short, direct and with few joints; in the latter, because of the greater length of pipe and larger number of joints, there is more opportunity for air infiltration and more resistance to flow, but larger and more economical auxiliaries may be used.

In order to permit of operating noncondensing when the condensing apparatus is out of commission, the exhaust pipe should contain a valve which can be opened to the atmosphere. This valve is usually arranged to open automatically when the condenser ceases to operate. To permit of repairs while the engine is running there should be a shut-off valve in the exhaust pipe leading to the condenser; and should several condensers discharge to a common main there should also be shut-off valves between them and that pipe.

319. Contact Condensers. (a) There are several different kinds of contact condensers only a few of which will be described. That in Fig. 429 is known as the "Ordinary *Jet Condenser*." * In it the injection water entering at I and the steam entering at S mingle in the conical condenser head B and the resulting mixture of condensate, injection water and noncondensable gases is raised to atmospheric pressure and discharged by the wet air pump located below, the flow of injection water being regulated by handwheel H. At (a) in Fig. 429 is a diagram of the piping for such a condenser. It includes an atmospheric relief valve (A) which will automatically open to the atmosphere when the valve V is closed for making repairs to the condenser, or when the vacuum is "broken," as when the injection water fails.

* The term "Jet Condenser" is also used as being synonymous with "direct contact condenser."

If the suction lift for the injection water is not too great this water may be siphoned into the condenser by the vacuum after it has been established by priming and starting the pump. In such cases this lift may be as much as 15 to 18 feet, provided the piping is short and not restricted. When the water is supplied in this manner there is danger of flooding and wrecking the engine in case the pump ceases to operate before valve *C* is closed, or if it runs so slowly that it cannot discharge the water as fast as it collects. To prevent the possibility of such disaster various

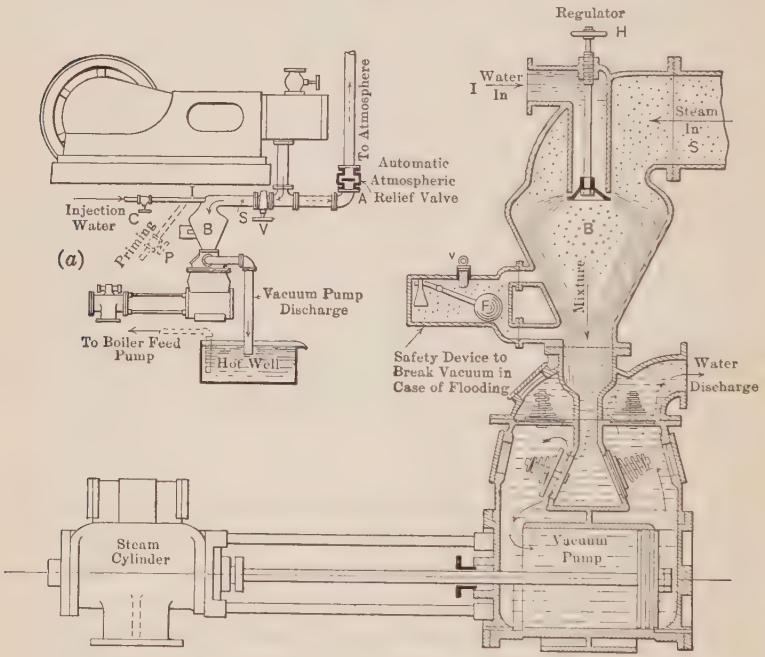


Fig. 429. — Jet Condenser.

expedients are adopted, such as providing a float (*F* in Fig. 429) which, when the water level becomes dangerously high, will be raised and open a valve *v* to admit the atmosphere to the condenser and thus "break" the vacuum and stop the flow of injection water.

If a pump is used for the injection water the head against which it operates is the difference between the total head and that through which the water would be "drawn" by the vacuum.

(b) The term "*Barometric Condenser*" may be applied to any form of direct-contact condenser having the barometric tube. Fig. 430 shows one arrangement commonly called the "*Siphon*" *Type*.* The injection water entering the condenser head *B* from pipe *I* passes downward in a thin annular sheet around the hollow cone in the condenser head and unites with the steam which passes through the cone. The mixture is discharged through the neck or throat *N* with sufficient velocity to carry with it the noncondensable gases. *A* is an atmospheric relief valve and *H* is a handwheel for regulating the injection water. The water level *L* in the tail pipe depends on the vacuum maintained, but for safety the tail pipe is extended 34 feet above the water level in the hot well.

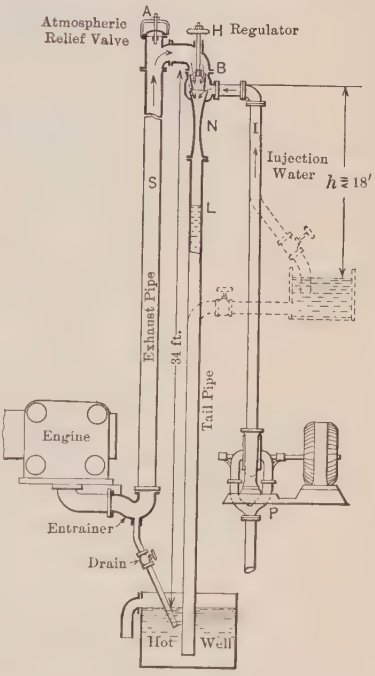


Fig. 430. Siphon Condenser.

If injection water is available at a head *h* (in the figure) of not over 18 feet, it may be "drawn in" by the vacuum after this has once been established by opening the lower valve shown dotted, or in some other manner, and in such case, the pump *P* can be dispensed with. At the foot of the exhaust pipe there should be either a drain, or an "entrainer," the latter being so arranged that the exhaust steam impinges on the surface of the water which has collected in a pocket, and gradually picks it up in small particles and disposes of it by entrainment. Because of the great head room required by these condensers they are frequently located outside of the power house; and sometimes a Tail Pump is substituted for the tail pipe, as in Fig. 431.

(c) Fig. 431 shows a direct-contact condenser somewhat simi-

* All direct-contact condensers can be used to siphon the condensing water, but the term "Siphon Condenser" is generally applied only when there is the neck *N* shown in Fig. 430.

lar to the one just discussed except that it uses a "*dry air pump*" for removing the air. As the volume of air to be handled will increase with its temperature, and as the size of the dry air pump will increase with the volume of the air, the latter is usually cooled in some manner before it goes to the air pump. In the arrangement shown this is done by passing it through a spray of cold water, in the upper part of the condenser head, on its way to the discharge opening.

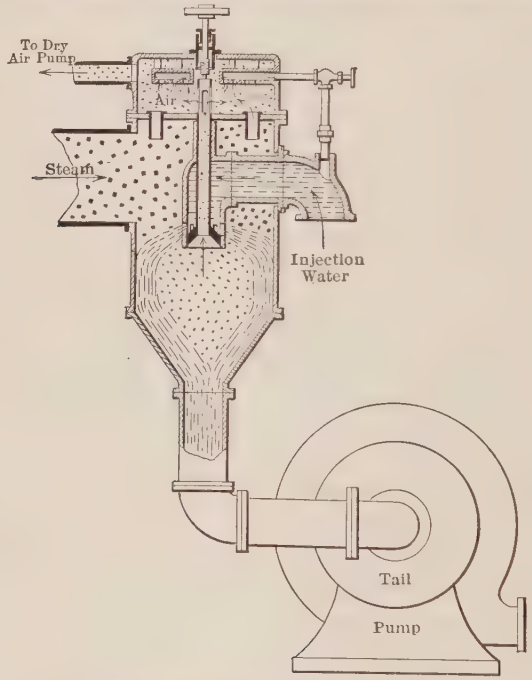


Fig. 431. — Condenser with Dry Air Pump.

(d) The *Ejector Condenser*, shown diagrammatically in Fig. 432, operates on the same principle as the steam ejector which is used for forcing water into boilers against the pressure of the steam. The injection water enters at *I* and passes through the neck of the combining tube *B*, where it rapidly condenses the exhaust steam which passes through small nozzles in the wall of this tube. Some of the heat surrendered by the condensed vapor is converted into kinetic energy of the steam jets flowing through these nozzles and the momentum acquired propels the water

with high velocity through the neck. This velocity is reduced in the expanding tube below so that the pressure is raised to atmospheric when the end E is reached.

To start the flow of injection water with the arrangement shown, boiler steam may be admitted through the starting valve C. This steam then issues through the check valve D and partly exhausts the atmosphere from the injection pipe, thus causing the water to rise and enter the condenser. The valve

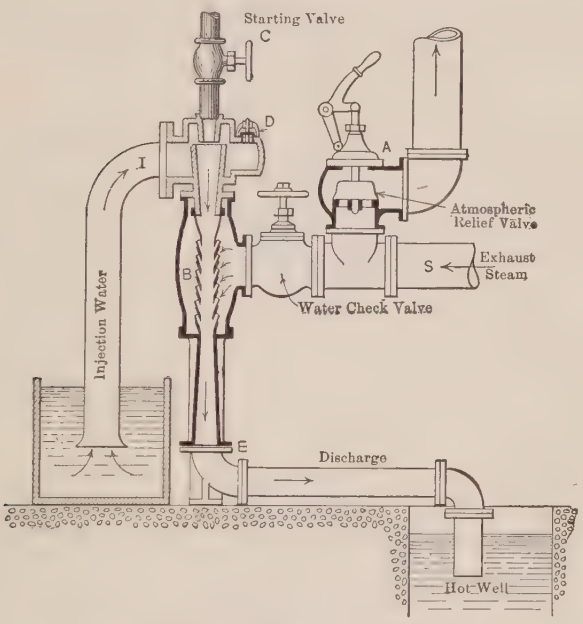


Fig. 432. — Ejector Condenser.

C may then be closed, the exhaust steam continuing the circulation of the water in the manner just described. The siphoning of injection water can also be started by admitting high pressure water through the starting valve C, in which case valve D can be omitted.

The operation of the condenser ceases, of course, when the supply of exhaust steam is discontinued, hence this arrangement of condenser cannot be used for intermittent service, nor is it satisfactory if the load varies widely and frequently. With steady load the "suction lift" may be 16 feet; with variable loads it is limited to a smaller value.

The combining tube may be arranged with adjustable internal throttling device and external sleeve to permit the regulation of the water and steam openings to suit the load. Should the water contain foreign matter a strainer should be located in the injection pipe.

If the condensing water is supplied under a head of 20 feet, or more, a slightly modified arrangement can be used and a more certain vacuum obtained even with wide variations in load.

(e) With all types of contact condensers the weight of water required per pound of *dry* steam for any vacuum is

$$w = \frac{\lambda_x - q_m}{(q_m - q_i)} = \frac{\lambda_x - (t_m - 32)}{(t_m - t_i)} * \quad . \quad . \quad . \quad (486)$$

where λ_x = total heat above 32° F. per pound of steam at exhaust pressure.

q_i = heat of liquid of injection water, at temp. t_i° F.

q_m = heat of liquid of mixture at temp. t_m° F. (t_m is from 5 to 15° less than the temperature of the exhaust steam.)

The temperature of the water in the hot well is practically that of the mixture, and this water is available for boiler feed when the character of condensing water permits. The weight of water to be handled by the circulating pump per hour is $w \times$ weight of steam condensed in that time, and the weight delivered by the discharge pump is $(w + 1) \times$ wt. of steam.

(f) The principal advantages of direct-contact condensers are: (1) Their simplicity; (2) low first cost; (3) low cost of upkeep; and (4) small space required. They have, however, certain detrimental features which in some instances may partly or wholly counterbalance these advantages: (*a*) If the injection water is sea water, or has scale-forming impurities, or is otherwise unsuitable for boiler feed, none of the heat in the condenser discharge can be returned to the boiler; (*b*) the distilled water resulting from the condensation of the steam is lost since it is mixed with the injection water, whereas with surface condensers it is available for boiler feed; (*c*) the temperature of the hot-well water used for boiler feed is lower than that from a surface condenser of proper design; (*d*) it is more difficult to obtain a good vacuum than with surface condensers, because of

* A correction of from 5 to 15 per cent must be made to allow for cooling the air and entrained moisture and for the inefficient heat absorption.

the air introduced by the injection water; and (*e*) larger air pumps are therefore required.

320. Surface Condensers. (**a**) A water-cooled surface condenser is essentially an enlargement in the exhaust piping through which pass tubes which contain the flowing condensing water. If this water flows merely from one tube header to the other, the apparatus is called a "single pass" condenser, and "multipass condenser" is the general term applied when the water flows across the steam chamber two or more times. A double-pass condenser of the ordinary type is shown in Fig. 433, with cooling

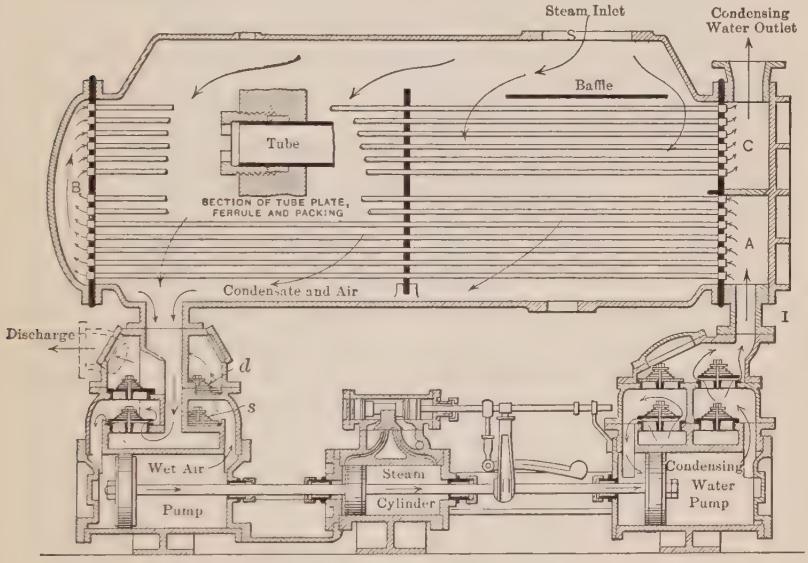


Fig. 433. — Double Flow Surface Condenser.

water flowing from the lower part (*A*) of one head to the other head (*B*) and then back to the upper part (*C*) of the first one.

The arrangement of piping for a surface condenser resembles that for the jet condenser in Fig. 429(*a*). In order to insure the flooding of all the condenser tubes at all times the condensing water is usually introduced at the bottom of the condenser and discharged at the top.

(**b**) The surface condenser has certain *advantages* over the direct-contact type. The principal ones are as follows: If the condensate is used as boiler feed, (1) substantially all of the available

sensible heat of the exhaust steam is returned to the boiler; (2) the same water is used repeatedly, thus avoiding the expense for new water (which is of importance only when suitable water is difficult to obtain or when its cost is high); (3) the feed water is distilled and free from scale-forming impurities; (4) less air is carried into the boiler by the feed water; and (5) sea water or any other water which is unsuitable for boiler feed can be used for cooling and yet the available sensible heat of the exhaust steam is returnable to the boiler; (6) better vacuums are generally obtainable with smaller air pumps and less power for same, because of (4) and because the air entrained in the condensing water is kept separated from the steam; and (7) there is no possibility of the circulating water flooding and wrecking the prime mover.

The principal *disadvantages* are the relatively large (*a*) first cost, (*b*) space occupied, (*c*) upkeep expense (the latter being largely due to the corrosion and deterioration of condenser tubes and to the multitudinous joints which must be maintained free from leakage), and (*d*), in the case of steam engines, the presence of oil in the condensate. The latter item does not hold with turbines. The surface condenser requires at least two pumps (the wet air and the circulating water pumps) and may use a third (a separate dry air pump) when the best results are desired. In contrast, some direct-contact condensers have no pumps, and others only a wet or dry air pump.

(c) The *weight of condensing water required* per pound of exhaust (with quality unity) is evidently

$$w = \frac{\lambda_x - q_c}{q_d - q_i} = \frac{\lambda_x - (t_c - 32)}{(t_d - t_i)} * \quad . \quad . \quad . \quad . \quad (487)$$

where λ_x = total heat above 32° F. per lb. of exhaust steam.
q_c = heat of liquid of condensate leaving condenser at t_c°.
t_c = from 0° to 20° F. below the exhaust temperature t_x.
q_i = heat of liquid of condensing water at inlet, at t_i.
q_d = heat of liquid of condensing water at discharge, at t_d.
t_d = from 5° to 10° F. below exhaust temperature t_x.

With from 25 to 26 inches of vacuum w is from 25 to 30 lbs. depending on the value of t_i; and with better vacuums w is from 45 to 55 lbs. and even more.

(d) The heat transmission in Surface Condensers is according

* See footnote on page 672.

to Case I of Sections 306 and 307 and the amount of condensing surface required to condense w_s lbs. of exhaust steam per hour (quality unity) is from Eq. (437).

$$S = \Delta Q \div \theta_m K$$
$$= w_s\{\lambda_x - (t_c - 32)\} \div \theta_m K, \quad . \quad . \quad . \quad . \quad (488)$$

where the symbols in the bracket have the same meaning as in Eq. (487); and from Eq. (438)

$$\theta_m = (t_d - t_i) \div \left(\log_e \frac{t_x - t_i}{t_x - t_c}\right). \quad . \quad . \quad . \quad . \quad (489)$$

The value of K depends on the surface coating on the tubes (scale and oil), on the velocity of the water, on the air present in the steam, on the material of the tubes (although this is usually negligible) and on other items. It ordinarily ranges from 250 to 300 B.t.u. per square foot per degree F. per hour in the simpler types of condensers under ordinary conditions, but with the best designs, well drained, and with good air pumps, the rate of transmission may be from two to three times these values. For ordinary condensers with from 24 to 26 inches vacuum about 10 lbs. of steam are condensed per square foot of heating surface per hour.

For small turbines with high vacuums from $2\frac{1}{2}$ to 4 square feet of condensing surface are ordinarily used per kilowatt rating of the generator; and with large turbines from 1 to $2\frac{1}{2}$ sq. ft. are found with the best types of condensers.

(e) The *essentials* which make Surface Condensers most effective are: (1) All the tube surface should be available for heat transmission; none of it should be air-bound either on the steam or water side. (2) The falling condensate should not "drown" any tubes, for then (*a*) the surface is only about 10 per cent as effective, (*b*) the condensate is cooler, hence not so valuable as feed water, and (*c*) more condensing water is required. (3) The velocities of the steam and water should be high enough to break up the surface films. As the transmission is largely dependent on the heat-absorbing ability of the water, the more rapidly the latter is brought in contact with the tube surfaces, the greater the rate of transmission. (4) The air in

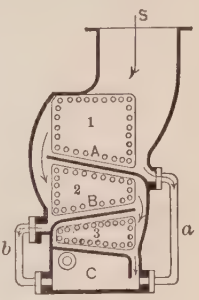


Fig. 434.

the condensate should be cooled as much as possible to decrease the volume to be handled by the air pump and to reduce its partial pressure acting on the prime mover. (5) There should be suitably arranged baffles to so distribute the steam that all parts of the condensing surface are equally effective.

(f) Fig. 434 shows one form of *dry-tube* condenser with arrangement for preventing the lower tubes of the condenser from

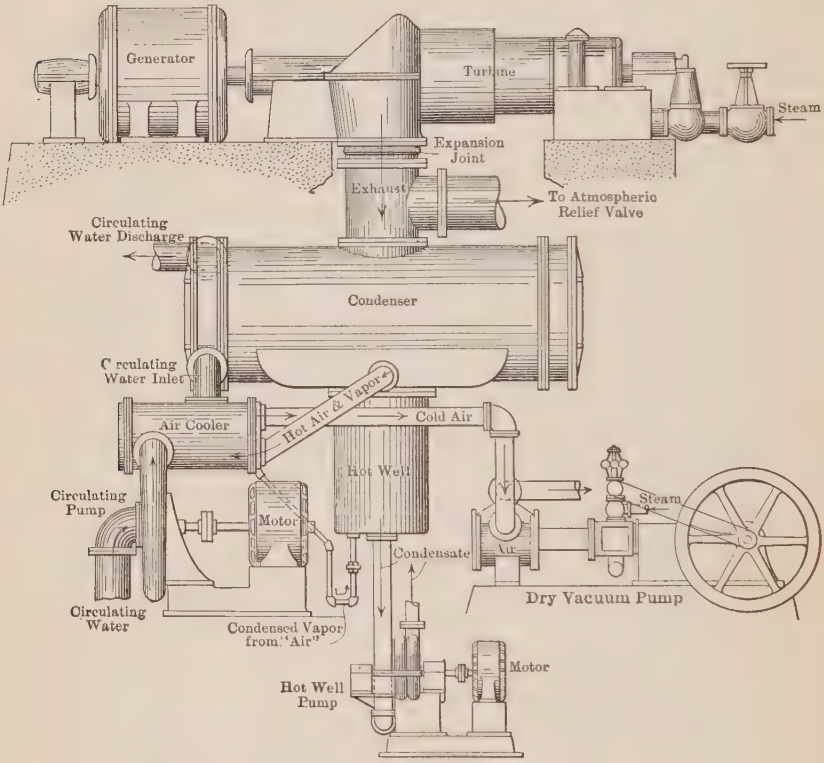


Fig. 435.

being drowned and from serving as condensate coolers. This is accomplished by the baffles A, B, which are arranged to collect and draw off the condensate from the tubes immediately above as rapidly as it is formed. Baffle pans somewhat similarly arranged (with drains) are used in like manner in other condensers of this type.

(g) In some condensers the counter-current principle is used

as regards the condensate. In such cases the exhaust steam enters the condenser at the bottom, hence the falling condensate passes downward through this upflowing steam and becomes heated thereby, the feed water then being substantially at exhaust temperature.

(h) As a cubic foot of air is heavier than a like volume of steam at the temperatures existing in condensers, and as it is coolest and most dense at the bottom of the shell, it tends to gravitate, hence the wet air pump placed below the condenser is correctly located for receiving the air as well as the condensate.

(i) Fig. 435 shows the piping of a condenser having separate dry air and hot-well pumps. The arrangement includes an air cooler through which the air passes on its way to the dry air pump, the condensed vapor from this cooler being passed through a water seal to the hot well where the condensate collects.

321. Air Pumps. (a) As has already been seen a *Dry Air Pump* is an air compressor which receives the air (and its entrained moisture) at condenser pressure and compresses it sufficiently to permit of discharge to the atmosphere, the compressor card resembling Fig. 436 in the best instances. As the compression ratio is high, the clearance volume must be small, for no air can be received from the condenser until that in the clearance space is expanded to condenser pressure. To increase the volumetric efficiency the three expedients most commonly used are: (1) The air may be compressed in two stages; (2) an equalizing passage (as *a* in Fig. 437) may be so arranged that at the end of the stroke the clearance air may at the proper time be momentarily exhausted into the other end of the cylinder which is filled with air at condenser pressure, thus when the stroke begins the clearance space is under pressure nearly equal to the vacuum; and (3) the clearance space may be filled with water, as is the case in wet air pumps.

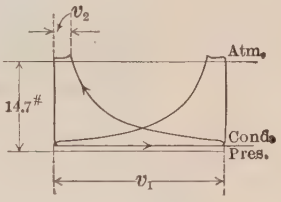


Fig. 436.

Air pumps as well as the other pumps are usually steam driven, as the exhaust steam can ordinarily be advantageously used for feed-water heating.

(b) A *Wet Air Pump* of the ordinary horizontal reciprocating type is shown at the left in Fig. 433, *s* indicating the suction valves and *d* the discharge valves. One of the vertical types is illustrated in Fig. 438 with foot valves, bucket valves and discharge valves as shown. In the *Edwards* type, shown in Fig. 439, the foot and bucket valves are dispensed with, and whatever condensate collects in the base of the pump is displaced and forced into the pump cylinder by the conical end of the plunger as it approaches the bottom of its stroke. This water and the air above are then caught above the plunger when it ascends and are discharged in the usual manner. As in most pumps, there is a lip around the upper valve deck so that the valves will always

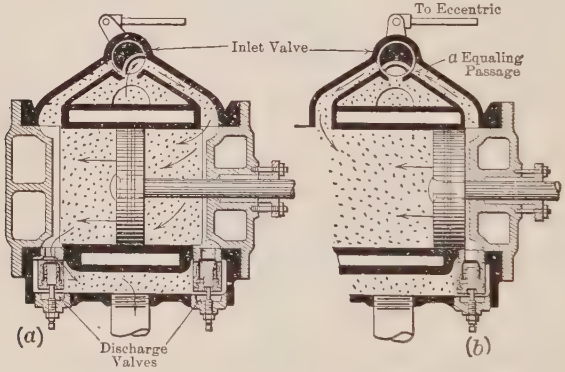


Fig. 437. — Dry Air Pump with Equalizing Passage.

be water sealed to prevent air leakage. Other single-acting vertical pumps and double-acting horizontal ones are somewhat similarly arranged to operate without foot and bucket valves.

A wet air pump of the *Rotary Type* is shown in Fig. 440, it being so arranged that the water-sealed lobed wheels not only discharge the water but also carry along the air which is introduced at *A* below the wheels.

(c) *The Leblanc type* of dry air pump is shown in Fig. 441. Water in chamber *A*, "drawn in" by the vacuum, is discharged at *B* and is projected downward by the vanes on the rotor *R* in a series of layers which, acting as pistons, entrap the air entering at *C* and force it through the neck *N* against the atmospheric pressure. To start the vacuum live steam may be temporarily admitted at *S*.

(d) The volume of condensate (V_c) and that of condensing water (V_w) used per minute can be readily estimated and the necessary *size of pumps* can then be determined if water alone is to be handled. If, however, air is to be pumped there is no exact method of arriving at the volume (V_a) to be discharged per minute and the proportions of the pump are based on rough

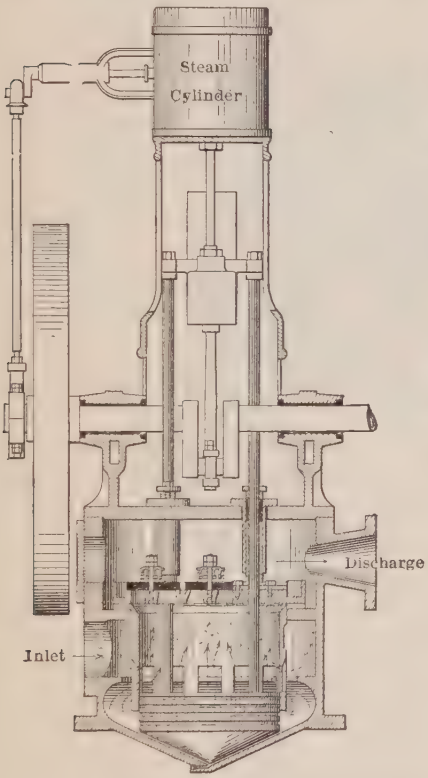


Fig. 439. — Wet Air Pump, Edwards Type.

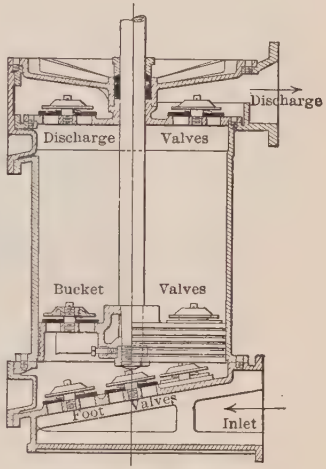


Fig. 438. — Wet Air Pump.

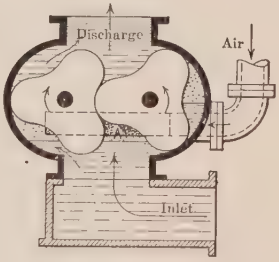


Fig. 440. — Rotary Type of Air Pump.

estimates. Surface water under atmospheric conditions may contain by volume from 2 to 5 per cent of air, and the leakages may increase the percentage of "atmospheric air" in the condensers to from 7 to 10 per cent. Under partial pressure and higher temperature conditions the volume of this air is greatly increased and its value can be readily computed, and serves as a rough basis for determining the size of the pump.

According to Gebhardt* single-acting wet vacuum pumps for *jet condensers* ordinarily have plunger displacements per minute of about 3 V_w, where V_w is the volume of the injection water pumped in that time, and double-acting pumps have displacement $3\frac{1}{2}$ V_w, the piston speeds being about 50 feet per minute. With reciprocating engines the *wet air pump for a surface con-*

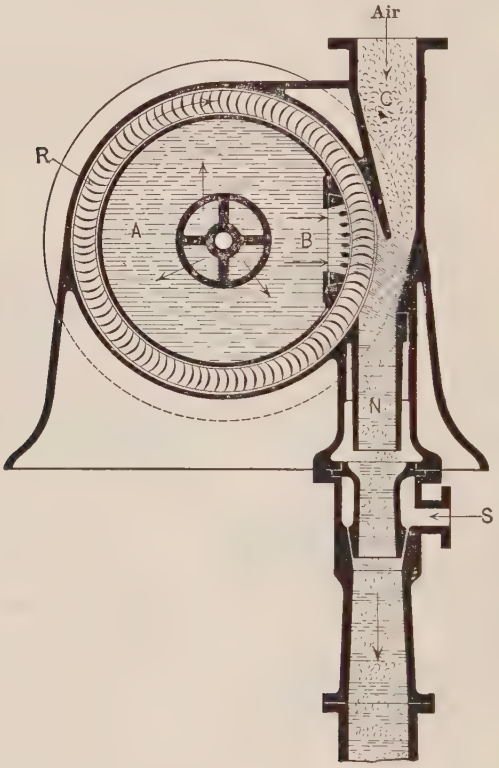


Fig. 441. —Leblanc Air Pump.

denser ordinarily has a displacement of 10 V_c, where V_c is the volume of condensate, and for steam turbines it is about 20 V_c, these values being the average of two hundred plants. For *dry air pumps* the displacement of the plunger ranges from 20 to 30 V_c with vacuum below 27 inches, up to 50 V_c for 28 inches and over, these values being based on an investigation of fifty installations.

* Gebhardt's "Steam Power Plant Engineering," published by John Wiley and Sons.

322. Recovery of Condensing Water. (a) The amount of water required in a plant for condensing purposes is relatively very great, varying, as shown in Sec. 320(c), from about 25 to 55 pounds per pound of steam condensed. After being used this water is generally wasted, hence a continuous supply of fresh water is required in such cases. When a plant owns its own water supply or is situated near a large river, or other body of water, from which it can pump condensing water, the cost of the water is practically only that of pumping. Many plants are so situated, however, that the only source is the city mains and in such cases the continuous expenditure for condensing water may be far in excess of the saving effected by its use. Methods of cooling and storing condensing water have been developed, therefore, so that the same water can be used repeatedly and thus make it possible to obtain the benefit of condensing operation in cases where the cost of a continuous supply of fresh water would be prohibitive.

(b) For cooling the water, various evaporative cooling devices are in use. They all operate by exposing large surface of water (sometimes in thin sheets or in drops) to air currents, the cooling being effected both by the direct contact of the cooler air with the hot water and by the evaporation of part of the water. For this purpose (1) a pond having relatively large exposed surface may be used; or (2) the water may be sprayed into the air and allowed to fall into a pond; or, (3) it may be passed through a cooling tower, such as described in the following paragraphs.

(c) *Cooling towers* are roughly divided into two classes:

1. Natural draft cooling towers, and
2. Fan towers or forced draft towers.

In the *natural draft type*, a vertical, rectangular or cylindrical shell is filled with some material or structure (trays, slats, wire screens, etc.) adapted to spread the water into thin sheets or streams. The water is introduced at the top, gravitates over this filling to a reservoir in the base of the tower and is then returned to the condenser.

Air enters at the bottom of the tower and passes upward through the filling so that cooling takes place on the counterflow principle. The upward motion of the air in the tower is due to the fact that its temperature and humidity are greater than

those of the outside air and it is therefore constantly displaced upward by fresh, cool air entering at the bottom. This effect (the "draft") is increased by lengthening the column of hot moist air by the addition of a "flue," or "stack," above the filling.

(d) A *fan type* of tower is essentially the same as one with natural draft so far as filling and cooling are concerned; the stack is omitted, however, and the draft is assisted by fans

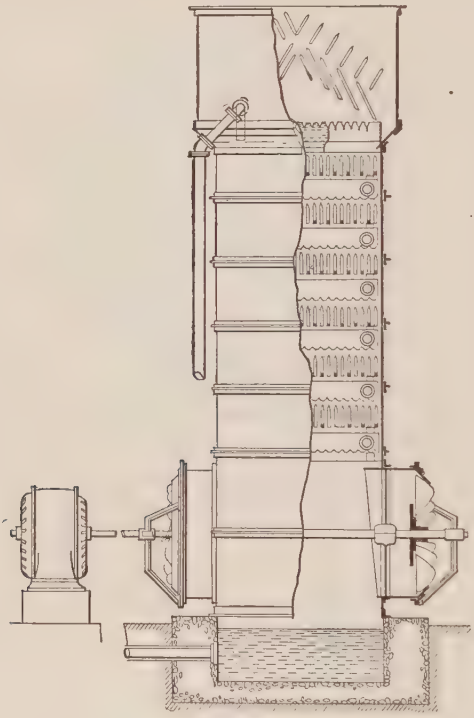


Fig. 442.

which force air in at the bottom of the tower. Such a structure, of which there are many arrangements, is shown in Fig. 442. This type of tower has the disadvantage of requiring an expenditure of power to operate the fans, but is independent of atmospheric conditions so far as draft is concerned.

A combination of both types is occasionally used, the stack supplying draft when possible and being helped out by the fans when necessary.

(e) Neglecting losses, the heat abstracted in a given time from the exhaust steam in condensing it equals the weight of the condensate (w_x) times the latent heat (r_x) of the exhaust steam at its partial pressure, and this, of course, is the amount of heat absorbed by the cooling water. Hence, if this water is used repeatedly, it must first be cooled by the surrender of this same amount of heat before its return to the condenser. This cooling is accomplished principally by the evaporation of a portion of the water, the heat carried away in this manner being equal to the product of weight (w_v) vaporized in the given time, by the latent heat (r_v) at the partial pressure existing at the surface exposed to the atmosphere. Then, considering that the cooling is effected entirely by evaporation and neglecting losses, it follows that

$$(wr)_x = (wr)_v; \quad . \quad . \quad . \quad . \quad . \quad . \quad (492)$$

or, the weight of cooling water evaporated in a given time is approximately

$$w_v = w_x r_x / r_v. \quad . \quad . \quad . \quad . \quad . \quad . \quad (493)$$

But, as r_x and r_v do not differ greatly, it is roughly true that $w_v = w_x$, — that is, under the conditions assumed, the weight of condensing water vaporized in the cooling device is about equal to the amount of condensate formed in the same interval of time in the condenser in which that water is used. Thus, if all the steam generated is condensed in a surface condenser and returned to the boiler, no new water (theoretically) is needed for boiler feed, but about an equal weight of make-up water must be constantly added to the supply of cooling water; and with surface condensers this water may, of course, be of quality unsuitable for use in the boilers. This is theoretically the maximum amount that need be lost in the process of cooling.

(f) If the air were so fully saturated that it could receive no more moisture, none of the water would vaporize and no cooling would be effected in the manner just described. In such case heat could still be abstracted from the water by bringing cooler air and its moisture in contact with it. The cooling media would then have their sensible heat raised by absorbing heat from the water, but it would take a great quantity of air to effect the cooling in this manner.

(g) The actual case is intermediate between the two extremes just discussed. The atmospheric air is practically never fully

saturated but nearly always has some humidity. In the average cooling tower from $\frac{1}{3}$ to $\frac{1}{5}$ of the heat is carried away by the increase in the sensible heat of the air and its vapor, and the rest by evaporation. The actual operation of the cooling device is dependent on the humidity, temperature, amount and distribution of the air and on the temperature and extent of exposed surface of the water. Ordinarily, under unfavorable conditions, one cubic foot of air entering can be expected to remove at least $2\frac{1}{2}$ B.t.u. as sensible heat of the air and latent heat of vaporized water; from 2 to 4 per cent of the condensing water is all that need be lost by evaporation; and the condensing water can be readily cooled 40 to 50 Fahrenheit degrees.

CHAPTER XXXVIII.

WATER PURIFICATION

323. Impurities in Natural Waters. (a) Waters available for power plant use are never the simple H_2O of chemistry but always carry certain impurities in suspension and in solution. When taken from streams or lakes the water generally has large quantities of mud and silt in suspension at certain periods of the year; in some cases at all periods. Water taken from sources which receive large deposits of leaves, twigs and other vegetable and animal remains will always carry certain organic substances in solution and sometimes in suspension as well. Practically all waters found on, or below, the earth's surface contain inorganic salts and gases in solution and sometimes free acids as well.

(b) All such impurities are liable to cause trouble in power plants, either (1) by clogging tubes and pipes, or (2) by corroding metal surfaces, or (3) by incrusting heat-transmitting surfaces, or (4) by causing foaming within boilers and similar apparatus.

(c) When solid material in suspension is of large size it can often be separated by simple mechanical means, as by screening, by settling, or by filtering through beds of coke, broken rock, or sand. When fine it can be removed by first entangling it in a flocculent precipitate and then filtering, as is done with municipal supplies.

In general, however, after such treatment average water will still contain in solution quite a quantity of material which will cause trouble if allowed to enter the apparatus of a power plant. The greatest difficulty is experienced in steam boilers and in the jackets of internal-combustion engines because of the deposition of such dissolved material upon the metal surfaces, thus forming a crust, or a scale, which materially decreases heat conductivity, clogs the passages, and may lead to the overheating of metal plates or surfaces. Acids in solution may cause corrosion of such surfaces.

(d) To prevent troubles of this character water is often "treated," "softened," or "purified," before use. The problem of water treatment is very large and by no means entirely solved as yet. In the following paragraphs the fundamental principles and the most common methods of treatment of boiler feed waters will be very briefly considered.

324. Troubles from Untreated Feed Water. (a) If untreated water is fed to a boiler the following troubles may ensue:

(1) Corrosion may occur because of

(*a*) *Free acid*, such as H_2SO_4 and HNO_3, which will not only attack the metal of the boiler, but if present in larger quantities than 5 parts per 1,000,000 * will often corrode or pit the metal parts of the engine with which the steam comes in contact; and because of

(*b*) *Organic material*, such as infusions of leaves, sewage and such, which acts as though acids were formed within the boiler.

(2) *Incrustation* may occur because of

(*a*) The *deposition of suspended matter*, such as mud, in parts of the boiler in which the circulation is not sufficiently rapid to maintain it in suspension;

(*b*) The *concentration of salts* brought into the boiler with the feed water and left behind by the issuing steam;

(*c*) The *deposition of decomposed salts*, such as the soluble bicarbonate of calcium, $Ca(HCO_3)_2$, which when heated loses one molecule of CO_2 and one of H_2O, leaving insoluble $CaCO_3$;

(*d*) The *deposition by heating* of salts which are less soluble in hot water than in cold, as calcium sulphate, $CaSO_4$, the solubility of which at ordinary steam temperatures is only about one-fourth as great as at ordinary atmospheric temperatures; and

(*e*) The *deposition of soaps* formed by the saponification of greases and organic oils by alkalies present in the feed water.

* Parts per, 1000, 100,000 and 1,000,000 are the terms commonly used for expressing the results of water analyses. Since one U. S. gallon of water at 60 degrees F. weighs 58,335 grains, one part per 100,000 is equal to 0.584 grain per U. S. gallon.

(3) *Foaming* may be caused by

(*a*) The decomposition or modification of salts to form flocculent precipitates which collect as a scum on and near the surface of the water; and by

(*b*) Organic matter, grease, soaps and such, which form similar scums.

(b) By far the most troublesome salts commonly found in feed waters are those of calcium and magnesium. They are generally either the carbonates or the sulphates of these metals. The carbonates of calcium form a more or less granular scale which it is not very difficult to remove with the tools used for cleaning boilers. The sulphate of calcium forms a very hard, porcelain-like scale which is removed only with great difficulty. Magnesium carbonate generally gives rise to a scum, causing priming, and also forms a hydrate which serves to cement together other scale-forming materials. The sulphate of this metal decomposes at high temperatures, liberating sulphuric acid which may cause corrosion and forming the hydrate which acts as a cement.

325. Methods of Treating Feed Waters. (a) A large quantity of the material carried in solution in boiler feed waters can often be precipitated by simply raising the temperature. This should be done in open type heaters using exhaust, or live steam, whichever is necessary to attain the necessary temperature. The impurities which are precipitated either remain fastened to the pans and other parts of the heater or are separated by filtering through a bed of coke, or other material, contained within the heater itself.

(b) Where the use of live steam is not desirable or where the water contains salts that are not readily precipitated by simply raising the temperature, certain chemicals can be added to the water. These chemicals should be so chosen as to react with the majority, or with the most harmful, of the impurities to form insoluble precipitates, or to form less harmful, soluble compounds. By far the most common chemical in use for this purpose is soda ash (impure sodium carbonate) although various other inorganic and organic compounds are also utilized. Soda ash has the advantages of very low cost, small weight required,

applicability to most waters and formation of compounds which are easily disposed of in the heaters and in the boilers.

(c) In many cases, particularly where steam is not available, or where special conditions are to be met, cold methods are used. In such cases a solution of the proper chemical, or chemicals, is fed in measured quantities to the raw water, and any precipitates formed are settled or filtered out, after which the treated water passes to some sort of storage to await use.

Apparatus of this type is generally made wholly or partly automatic. It is always of large size and therefore costly, and because of the low temperature many reactions which may be easily carried out in heaters are either entirely absent or are very incomplete.

(d) Many "*boiler compounds*," some of secret composition, are in use. They are mixed with the feed water on its way to the boiler and are supposed to prevent or mitigate the formation of scale. It should be remembered that no solid material which enters the boiler can leave with the steam and hence it must all remain within the vessel unless removed by other means, such as blowing down, skimming, etc.

This being the case, all that can be expected of a boiler compound is that it will react with the most troublesome impurities so as to change them to less troublesome ones which can be removed as sludge through the blow-off valve, rather than as a hard scale adhering to the metallic surfaces. In any event the amount of solid to be removed from the boiler will be greater when a compound is used than when the untreated water is vaporized, and it is merely a question of whether the greater amount of soft material permits of more economical and safer operation than does the smaller quantity of harder scale. It is therefore obvious that the impurities should, when possible, be removed from the water *before* it is introduced into the boiler.

(e) There are some so-called "boiler compounds" on the market which are not supposed to react with the solids in the water but are intended to coat the water side of all heating surfaces in such a way as to prevent the adherence of scale and scale-forming material. Besides the commercial compounds, kerosene and similar hydrocarbon oils have been more or less successfully used for this purpose. Oil so used should contain

no organic admixture as this may cause trouble by saponifying in the boiler and it should contain no heavy hydrocarbons which will form tar or pitch as these might cause overheating of plates to which they become attached.

It has also been claimed that graphite acts in a way similar to kerosene in preventing the adherence of scale-forming material.

CHAPTER XXXIX.

POWER PLANTS.

326. General. (a) Only a very general discussion of the subject of power plants as a whole can be attempted in this book and that must be given in the briefest manner possible. Plants having internal combustion engines and those having steam-operated prime movers will be the only types considered.

(b) The choice between plants of these two types depends on many considerations, some of which are: (1) kind of fuel available, (2) fuel economy, (3) first cost and other financial considerations, (4) reliability, (5) weight, (6) space occupied, (7) cost of water, (8) ability to secure properly trained attendants, (9) location, and (10) character of load. In general, where coal is very expensive, the producer plant will give better financial returns than a steam plant unless the power requirements are such as to call for unusually large units (say, from 1000 to 4000 or 5000 horse power for the plant).

327. Internal Combustion Engine Plants. If the fuel is oil, or gas, the plant merely consists of the engine with means of supplying the fuel and for transmitting the energy developed, and with provision for jacket water. If solid fuel is used in a producer, the elements of the plant are those given in Fig. 443.

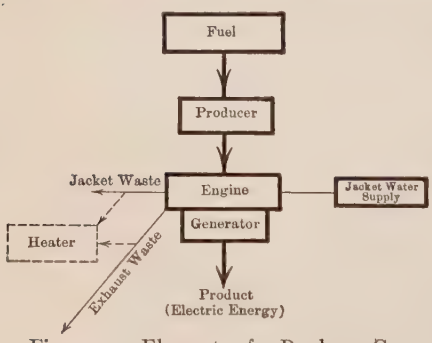


Fig. 443. — Elements of a Producer Gas Power Plant.

328. Steam Power Plants. (a) The *location* of the plant is selected with respect to (1) railroad and dock facilities for receiving coal and disposing of ashes, (2) supply of water suitable for feed and condensing purposes, (3) convenience for distribu-

tion of its products (electrical energy, exhaust steam for heating, belt delivered power, etc.), (4) cost of real estate, (5) suitability of ground for foundations, (6) space for storage of fuel, (7) character of the surrounding neighborhood, and (8) allowance for increase in size of plant.

(b) The *building* is generally divided by a fireproof and dust-

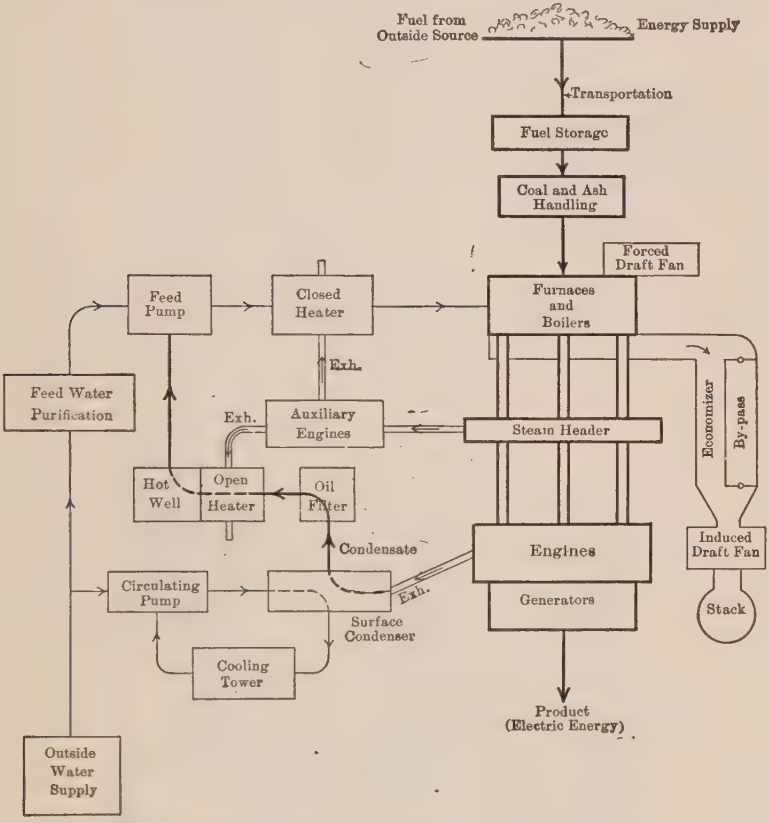


Fig. 444.—Elements of a Steam Power Plant.

proof wall into the boiler room and the engine (or turbine) room, and is provided with proper lighting and ventilating facilities and with doorways of sufficient size to admit the largest pieces of the equipment. In large plants the railroad track usually enters the building, the doors being large enough to admit box cars. The architecture of the building should be in harmony

with its surroundings and the design should, in general, permit of enlargement of plant to meet increases in the demand for power.

The scheme of the steam power plant equipment is illustrated in Fig. 444. This diagram is very general and includes pieces of equipment which are used only in special cases; it also shows apparatus which would not, in general, be used at the same time in ordinary cases. Several arrangements of steam power plants are shown in Figs. 445 to 449.

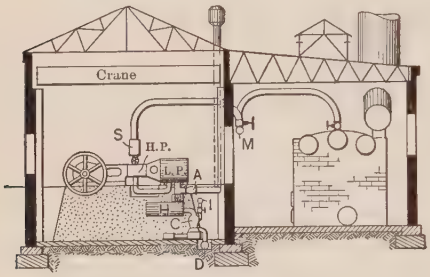


Fig. 445. — Small Engine Plant.

(c) In the *boiler room*, besides the boilers, are located the feed pumps, fans, feed heaters (generally), economizers, etc. Car tracks are arranged to deliver the coal at such point as to reduce the manual labor to the minimum. Large plants usually have overhead bunkers, to which the coal is brought by cars or by mechanical conveyors of the bucket, belt, or other type, and from which chutes lead to the hoppers of the stokers, or, in case of hand firing, to the floor in front of the boiler; and ash hoppers

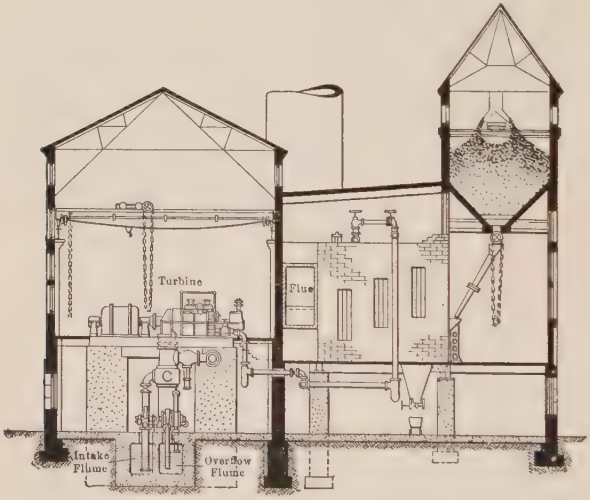


Fig. 446. — Small Turbine Plant.

are generally placed under the grates with dumps discharging to conveyors or cars below. In some instances boilers are located on two or more floors (as in "double deck" plants) as in

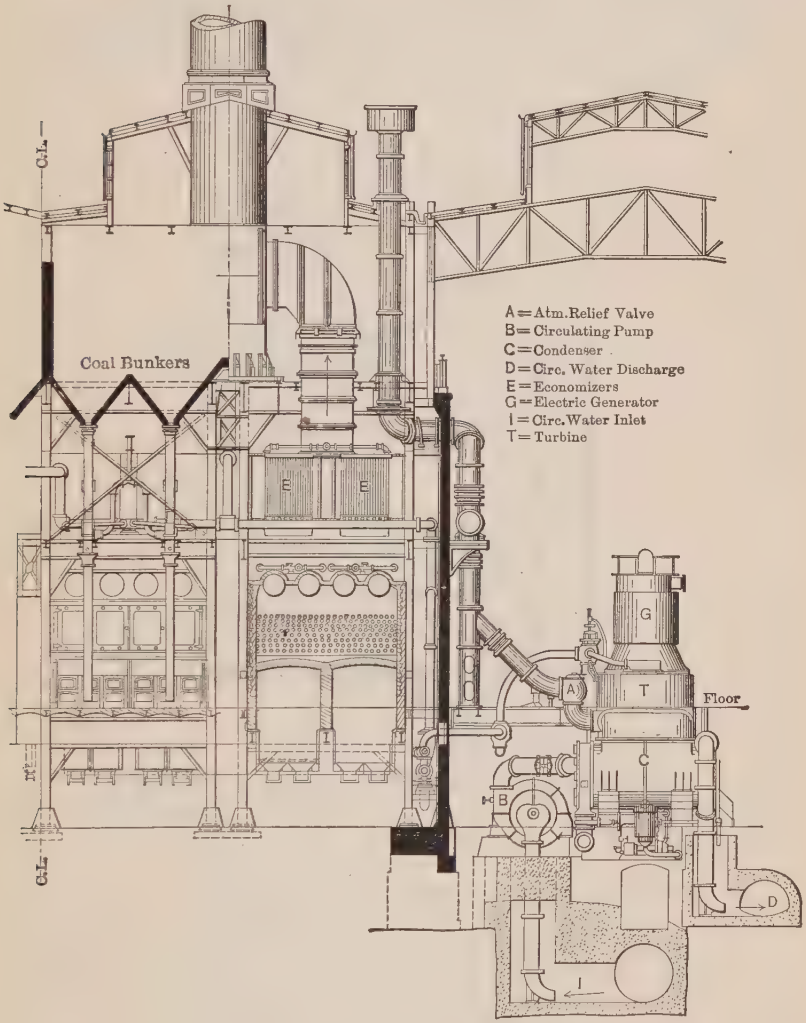


Fig. 447.—Large Turbine Plant.

Fig. 448, or are fired from both ends (Figs. 367 and 449), to reduce the ground area occupied. These special arrangements are more often adopted in turbine plants than in those having engines,

because the turbine room is generally much smaller than the boiler room, whereas an engine room is ordinarily of about the same size.

(d) The larger *engine rooms* are usually provided with overhead traveling cranes of capacity at least sufficient to lift the heaviest piece of machinery. Surface and jet condensers and their pumps are usually located below the engine, in the base-

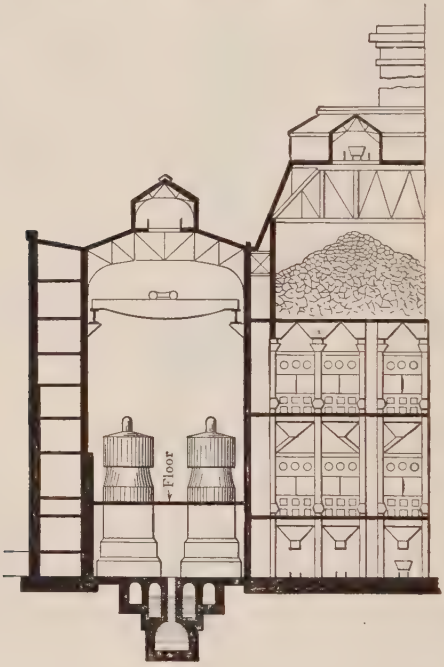


Fig. 448. — Power Plant with Double Deck Boiler Room.

ment; and barometric and siphon condensers are often placed outside the building.

(e) In electric plants as much of the *steam piping* as is feasible is located in the boiler room to prevent liability of damage to the electrical apparatus by the steam in case of pipe failure. Where plant shutdowns are of serious consequence, the ideal arrangement of piping would permit (1) of running any prime mover from any boiler, (2) of any boiler, or engine unit, being isolated without affecting the rest, and (3) of making repairs to any portion of the piping without affecting any unit (or not more than one). This ideal case is approximated most closely

by the "*loop or ring*" system of piping, *a* in Fig. 450, and by the "*double-header*" system, as *b* in Fig. 450. These systems call for an amount of piping and a number of joints and valves that is prohibitive in most cases. Ordinarily the connections from the boilers are merely led to a "*single header*" from which other pipes lead to the prime movers, as *c* in Fig. 450. In large plants it is common practice to arrange each prime mover and

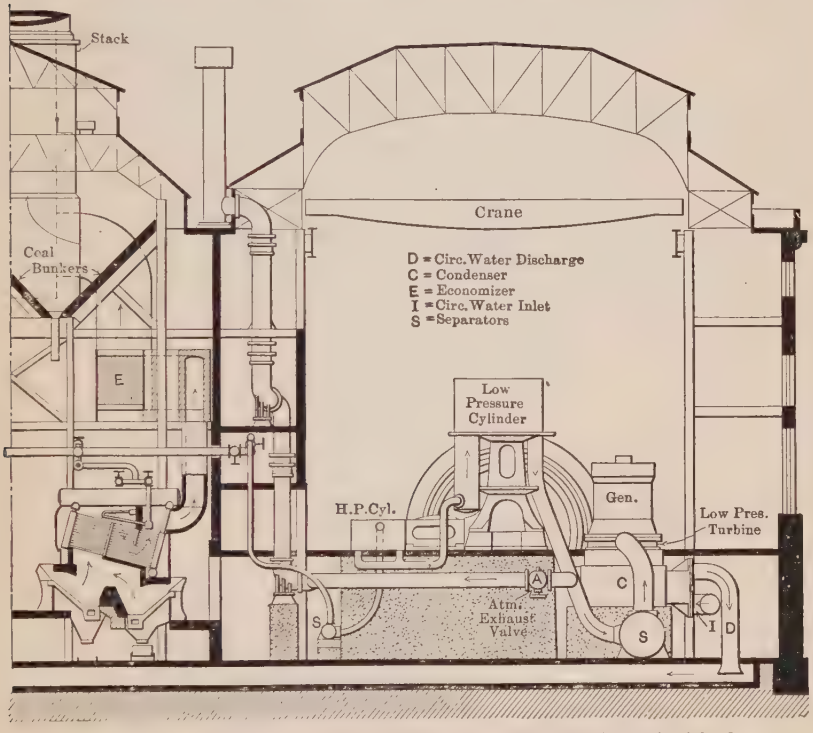


Fig. 449. — Power Plant with Boilers Fired from Both Ends, and with Compound Steam Engines Exhausting to Low Pressure Turbines.

the boilers which serve it, as an independent unit; but frequently cross-connections are provided between units for use in emergencies. There is almost an unlimited number of arrangements of piping possible, but they are generally modifications or combinations of those just given.

The piping for the auxiliary apparatus is independent of the main system to permit that apparatus to be operated even though the other is not in use.

The steam pipes from the boiler have hand-operated shut-off valves, and in some cases also include emergency valves which will close if abnormal outflow of steam occurs, as when a steam pipe is ruptured, and act as check valves preventing the inrush

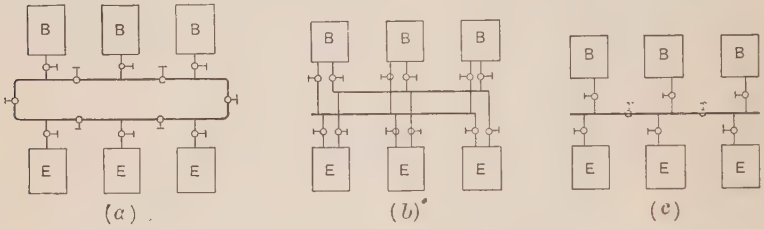


Fig. 450.—Steam Piping Arrangements.

of steam from other boilers if a tube fails. The engine feeders also have shut-off valves near the cylinders and sometimes there are also valves which will automatically close if the engine starts to run away. All steam piping should be lagged with non-

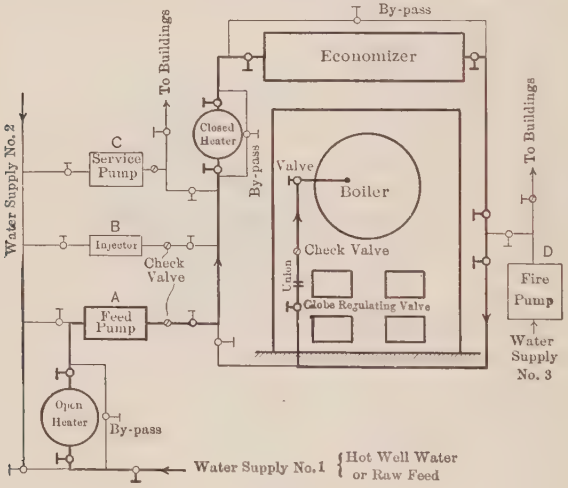


Fig. 451.

conducting covering; it must be properly supported; provision for expansion must be made by the introduction of slip or swivel joints, corrugated sections or flexible curved portions; and it must be so arranged as to be without undrained portions from which the collected water can be carried over in large quantities to the prime mover with disastrous results. The piping

should be properly drained by suitably arranged collecting pockets connected with "traps" or equivalent devices and at the engines there should be "steam separators." The traps (by float or other device) automatically discharge the accumulation of water from time to time.

(f) The *boiler feed-water piping* is preferably so arranged that any of several pumps (or injectors) can be used for supplying the feed water, and also so that there are several sources from which this water can be obtained. One of the numerous possible arrangements of piping is diagrammed in Fig. 451.

(g) Each piece of auxiliary apparatus is preferably so arranged that it can serve any one of the sets of main units, and so that it can be isolated if out of commission and the materials which it ordinarily handles may be by-passed around and led direct to their final destinations.

CHAPTER XL.

CONTINUOUS FLOW OF GASES AND VAPORS THROUGH ORIFICES AND NOZZLES.

329. Introductory. (a) The thermodynamic transformations previously discussed in this book were assumed to occur in such manner that the change of kinetic energy (velocity energy) associated with the flow of the working substance from one part of the system to another was zero. In the usual cases of flow of gases or vapors through pipes and in cylinders this same assumption may be made without introducing serious error, because of the relatively low velocities prevailing.

When, however, gases or vapors flow through nozzles or orifices both the changes in velocity and the corresponding changes in the kinetic energy of the working substance may be very large. In such instances much of the potential energy associated with the substance may be converted into the kinetic energy associated with the change of motion, or the reverse process may occur.

(b) Let Fig. 452 represent a conduit through which the material flows in the direction of the arrow; and let Po and K be respectively the simultaneous potential and kinetic energies associated with a pound of the working substance at any instant. Then $Po_1 + K_1$ is the total associated energy per pound when the stream passes section 1 in Fig. 452 and $Po_2 + K_2$ is its value when passing section 2. If the conduit is of such character that energy neither passes through, nor is stored by, its walls, it follows from the Law of Conservation of Energy, and from the First Law of Thermodynamics, that

Fig. 452.

$$Po_1 + K_1 = Po_2 + K_2, \quad . \quad . \quad . \quad . \quad . \quad (494)$$

from which

$$\Delta K = K_2 - K_1 = Po_1 - Po_2 \quad . \quad . \quad . \quad . \quad (495)$$

where ΔK is the Change in Kinetic Energy per pound of material. This equation shows that the changes in kinetic and

potential energies between sections 1 and 2 are equal in amounts but opposite in direction.

(c) For the purposes of analytical development it is desirable to examine more in detail the character of the potential energy which must be considered in connection with the flow of gases and vapors. One form of potential energy is that due to *position*, as exemplified by the familiar "head" in hydraulics. In most engineering problems dealing with the thermodynamics of flow of the materials under consideration the magnitude of this form is so small that it may be neglected without serious error. A second form is that due to a substance's *stored heat energy* which was expended in raising the sensible heat of the material, or in doing internal work, or in both; and it is obviously composed of the ΔS and ΔI which have been used before. The only other form of potential energy is that stored in the enveloping media due to *external work* done upon them by the substance surrounded. Although stored in the surrounding media, it is convenient to consider this potential energy as associated with the substance that did the work, because in all cases this energy would be returned if that substance were brought back to the original conditions. Evidently this stored external work corresponds to the ΔE used in previous discussions, but for subsequent purposes it will be convenient to obtain a slightly different viewpoint regarding it.

(d) For this purpose suppose the plug *a* (in Fig. 453), weighing one pound, and having a (specific) volume **V**, is injected its full length into a closed vessel *b* which is so large that the medium it contains may be assumed to remain at constant pressure P pounds per square foot during the process. Then, the external work done by the plug upon the medium is $P\mathbf{V}$ foot-pounds, or $P\mathbf{V}/778$ B.t.u. This energy is stored in the surrounding medium, but as it will be returned when the plug is withdrawn, it may be considered to be associated with the plug. Obviously all material, which is surrounded by media upon which it has done work which is stored as potential energy of this form, may be considered as having associated with it this returnable energy.

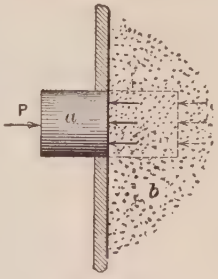


Fig. 453.

(e) Now, summing up, the total potential energy which may be considered as associated with a pound of the material is

$$Po = \Delta S + \Delta I + P\mathbf{V}/778 \quad . \quad . \quad . \quad . \quad (496)$$

measured above certain datum conditions which will be considered later.

This value of the potential energy may now be substituted in Eq. (494) giving, for the conditions of flow shown in Fig. 452,

$$(\Delta S_1 + \Delta I_1 + P_1\mathbf{V}_1/778) + K_1 = (\Delta S_2 + \Delta I_2 + P_2\mathbf{V}_2/778) + K_2, \quad (497)$$

and Eq. (495) then becomes

$$\Delta K = (K_2 - K_1)$$
$$= (\Delta S_1 + \Delta I_1 + P_1\mathbf{V}_1/778) - (\Delta S_2 + \Delta I_2 + P_2\mathbf{V}_2/778). \quad (498)$$

The change in kinetic energy ΔK, and hence the change in velocity, between sections 1 and 2 can thus be computed when the intrinsic energies $(\Delta S + \Delta I)$ and the $P\mathbf{V}$ conditions of the material as it passes these sections are known.

(f) If w pounds of substance are flowing per second with a velocity of v feet per second, the kinetic energy of the flowing material is $wK = wv^2 \div (2\,g \times 778)$, or the kinetic energy per pound is

$$K = \frac{v^2}{2\,g \times 778} \text{ B.t.u.} \quad . \quad . \quad . \quad . \quad . \quad (499)$$

If the velocity is increased from v_1 to v_2, the change in kinetic energy is, per pound of substance,

$$\Delta K = (K_2 - K_1) = \left(\frac{v_2^2}{2\,g} - \frac{v_1^2}{2\,g}\right)/778 \text{ B.t.u.} \quad . \quad (500)$$

But since $(K_2 - K_1) = (Po_1 - Po_2)$ this equation may be rewritten

$$\Delta K = Po_1 - Po_2 = \left(\frac{v_2^2}{2\,g} - \frac{v_1^2}{2\,g}\right)/778 \text{ B.t.u.}, \quad . \quad (501)$$

which may be used for determining the velocity changes when the changes in potential energy are known.

330. Flow of Saturated Steam in the Ideal Case. (a) Following the general equation (498), the kinetic energy change during flow, in terms of the change in potential energy, is

$$\Delta K = \left(\Delta S_1 + \Delta I_1 + \frac{P_1\mathbf{V}_1}{778}\right) - \left(\Delta S_2 + \Delta I_2 + \frac{P_2\mathbf{V}_2}{778}\right),$$

and as the right hand side of this equation represents differences between the various heat quantities, it is immaterial what is

taken as a datum. It is therefore convenient in the case of steam to use 32 degrees F. as such. With this assumption it is obvious that for saturated steam which is initially dry ΔS_1 and ΔS_2 must equal q_1 and q_2; ΔI_1 and ΔI_2 must equal ρ_1 and $x_2\rho_2$; and $P_1V_1/778$ and $P_2V_2/778$ correspond to $(APu)_1$ and $(xAPu)_2$ if the volume of water be neglected, which is permissible since it is relatively insignificantly small. The general equation for saturated steam may therefore be written in the form

$$\begin{aligned}\Delta K &= (q + \rho + APu)_1 - (q + x\rho + xAPu)_2 \\ &= (q + r)_1 - (q + xr)_2 \\ &= \Delta Q_1 - \Delta Q_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (502)\end{aligned}$$

in which ΔQ_1 and ΔQ_2 are the total heats above 32 degrees per pound of steam at points 1 and 2 respectively.

While the foregoing discussion considered only the case of dry saturated steam, it applies equally as well to any initial condition. Thus in the ideal case ΔQ_1 is the total stock of heat per pound of steam entering the nozzle and ΔQ_2 is that remaining when the lower pressure is reached by the process of expansion.

It is obvious that even though the initial and final pressures are known, Eq. (502) cannot be used for the solution of numerical problems until some means is found for determining the value of x_2, or ΔQ_2. But after they have been found ΔK can be determined; then from Eq. (501) the change in velocity of flow can be computed. Methods of determining the values of x_2 and ΔQ_2 will now be considered.

(b) Since the velocity of steam flowing through a nozzle or similar conduit is very high there is very short time of contact between steam and walls. And further, since the temperature of the steam in contact with any particular ring in the wall of the conduit is always the same so long as steady conditions are maintained, it follows that each part of the wall will acquire practically the same temperature as the steam in contact with it. As a result of these two facts there is in a real case very little transfer of heat between the walls and the steam and it is reasonable to assume no transfer in an ideal case, hence the flow may be considered an adiabatic process, or, since the pressure decreases as flow progresses, as an adiabatic expansion.

It can also be shown that for ideal conditions this adiabatic expansion may also be treated without serious error as isentropic, consequently all the mechanism previously developed for such conditions can be used in this case.

(c) Then, on the Tϕ-diagram in Fig. 454, if the initial state point is 1 (the condition being T_1, x_1, ΔQ_1), the ideal expansion in the nozzle would be along the isentropic line to point 2 where the condition is T_2, x_2, ΔQ_2. The heat ΔQ_1 per pound of steam at the beginning of expansion is shown by the area surrounded by the heavy line; and the heat ΔQ_2 at the end of the process, by the sectioned area; hence the net work ΔK done is $\Delta Q_1 - \Delta Q_2$ and is shown by area *ab*12. This area is seen to equal that surrounded by the lines of the Clausius cycle. Hence, in the ideal case *the change of kinetic energy* ΔK *occurring in a steam nozzle is equivalent to the external work,* $\Delta E = (\Delta Q_1 - \Delta Q_2)$, *done with a Clausius cycle having the same expansion line* and using the same weight of material.

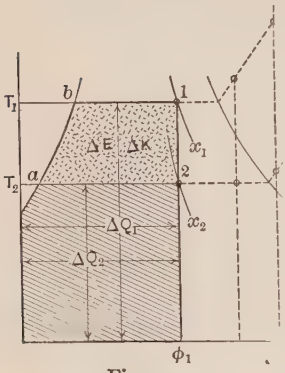


Fig. 454.

Thus ΔK $(= \Delta E = \Delta Q_1 - \Delta Q_2)$ can be computed by the methods given in Sect. 94 for determining the work of the Clausius cycle, or it can be obtained from the Tϕ-diagram or from the Mollier Chart.

(d) On the Mollier Chart, Fig. 455, the ideal (isentropic) process for one pound of steam is represented by the line 1–2. At the initial point the condition of the steam is x_1, p_1, ΔQ_1; and at point 2 it is x_2, p_2, ΔQ_2. During the process heat equal to $\Delta K = (\Delta Q_1 - \Delta Q_2)$, as shown by the length 1–2, is surrendered for producing the flow.*

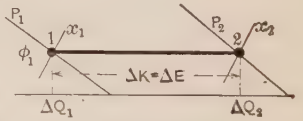


Fig. 455.

(e) Having found $\Delta K = (\Delta Q_1 - \Delta Q_2)$, the velocity of the stream passing point 2 is, from Eq. (501),

$$v_2 = \sqrt{v_1^2 + 50{,}103 \times Ef\,(\Delta Q_1 - \Delta Q_2)} \text{ ft. sec.} \quad . \quad . \quad . \quad (503)$$

in which Ef is the efficiency of heat conversion as compared with the ideal case. Its value is unity when no losses occur.

* The Ellenwood Chart (Appendix) is especially useful for nozzle problems as besides giving ΔK it gives values of $x\mathbf{V}$ used in Eqs. (507) and (508).

If the acceleration is from rest, or from a negligible initial velocity, which is ordinarily the case, then Eq. (503) becomes

$$v = 223.8\sqrt{Ef(\Delta Q_1 - \Delta Q_2)} \text{ ft. sec.}, \quad . \quad . \quad . \quad (504)$$

or

$$v = 223.8\sqrt{Ef \times \Delta K} \text{ ft. sec.} \quad . \quad . \quad . \quad . \quad . \quad (505)$$

(f) If the velocity of the stream is v feet per second at any section which has an area of a square inches, then the volume flowing per second through that section is $(va/144)$ cubic feet; also if w pounds of material are passing the section per second, and if the specific volume is $\mathbf{V}$* and the quality x, then the volume passing per second is $wx\mathbf{V}$. Hence, at the given section

$$wx\mathbf{V} = va/144 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (506)$$

Thus, if a given weight of vapor, of known specific volume and quality, is to be passed with a given velocity, the area of the passage must be

$$a = \frac{wx\mathbf{V}}{v} \times 144. \quad . \quad . \quad . \quad . \quad . \quad . \quad (507)$$

Or the weight of the material passed by a given area may be computed from

$$w = \frac{va}{144\,x\mathbf{V}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (508)$$

While the foregoing discussion was confined to steam, it, of course, applies equally well to any other vapor. But these equations cannot be applied indiscriminately as will be shown in the next section.

331. The Ideal Steam Nozzle. (a) Starting with any initial state $(p_1x_1\Delta Q_1)$ and expanding in a steam nozzle in the manner described, it will be found that as the terminal pressure is lowered certain peculiar phenomena occur which are difficult to understand without the aid of curves. These curves will now be constructed:

For a series of such expansions, all for one pound of steam and from a fixed initial absolute pressure p_1 and quality x_1, to

* The weight of the steam per pound of material is equal to the quality x and that of the moisture is $(1 - x)$; the volume of the steam is $x\mathbf{V}$ and that of the water is $(1 - x) \times$ volume of 1 pound of water. As the volume of one pound of water is about $\frac{1}{1600}$ that of the steam at atmospheric pressure and of like order at other pressures the volume occupied by the moisture in the steam is negligible and hence is not included in the discussion given above.

progressively decreasing absolute pressures p, let there be derived the successive values of:

(1) The quantities x at pressures p;

(2) The heat $\Delta K = \Delta Q_1 - \Delta Q$ theoretically made available for producing flow; and

(3) The specific volumes **V** at pressure p.

Items (1) and (2) may be readily obtained from the Mollier Chart or they may be computed by the methods given in Sect. 94; and (3) may be obtained from the steam tables. Then with the values of ΔK as abscissas plot curves, as in Fig. 456 (*a*), to show by ordinates how p, x and **V** change with the surrender of heat as the expansion progresses.

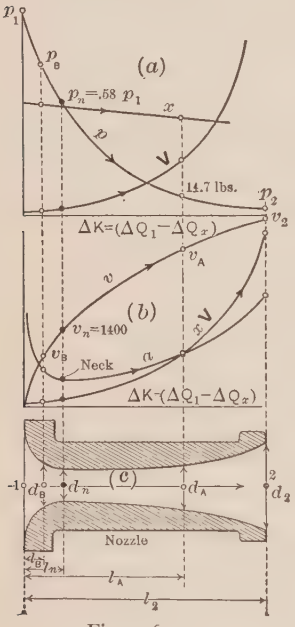


Fig. 456.

Next compute the progressive values of

(4) The actual volumes (x**V**) of one pound of material;

(5) The jet velocities v (by substituting the different values of $(\Delta Q_1 - \Delta Q)$, in Eq. (504)); and

(6) The areas a of sections required at different points along the nozzle to obtain these velocities. These areas may be obtained from Eq. (507).

Then with the same abscissas as before plot curves, as in Fig. 456 (*b*) to show by ordinates the variation of x**V**, v and a with ΔK.

(**b**) Now referring to the figure it will be seen from the a-curve that, as expansion progresses, the cross-sectional area of the passage must at first contract and then diverge, if expansion is carried far enough, — thus the nozzle must have a neck. Projecting upward from this neck to the p-curve, it will be found that the neck pressure is about 58 per cent of the initial pressure and that the corresponding point v_n on the v-curve will scale at a little over 1400 feet per second. And it is important to note that *about* these same values will always obtain regardless of

the initial condition of the steam,* hence they are called the *critical pressure* and *critical velocity*.

(c) Fig. 456 (*c*) shows the longitudinal section of a nozzle in which the amount of heat surrendered per inch of length is constant from one end to the other and is equal to the abscissa scale used for the curves above. This nozzle is for one pound of steam flowing per second, with terminal pressure p_2 pounds absolute and final velocity v_2, as shown on the respective curves. Corresponding to any other pressure of exit from the nozzle, the end diameter may be found by projecting downward from the proper point on the p-curve to the longitudinal section of the nozzle; — thus for discharge to atmospheric pressure (14.7 lbs.) the end diameter is seen to be d_A, the length of nozzle is l_A, and the terminal velocity is v_A; for discharge to a pressure $p_n = .58\ p_1$ the corresponding values would be d_n, l_n and v_n; and for terminal pressure equal to p_B they would be d_B, l_B and v_B. Evidently a nozzle originally used with discharge pressure p_2 will be theoretically correct for a case where the pressure is atmospheric, if its end is cut off sufficiently to have the final diameter beyond the neck equal to d_A, and similarly for the other terminal pressures. Thus regardless of the exit pressures, Fig. 456 (*c*) is the horizontal section of all nozzles discharging the same weight of steam per second with same initial conditions, the only difference between the various nozzles being in the lengths and end diameters, which are made to correspond to the terminal pressures.

(d) The reason the neck is present in all cases where the expansion is to a pressure below .58 p_1 can now be easily explained: From Eq. (507) the area of the nozzle at any section, per pound of material flowing per second, is

$$a = \frac{x\mathbf{V}}{v} \times 144.$$

Now referring to the curves for $x\mathbf{V}$ and v in Fig. 456 (*c*) it will be seen that as the expansion progresses the numerator $x\mathbf{V}$ at first increases much slower than does the denominator v, and hence the nozzle areas (a) must first diminish; but that later the conditions are reversed, consequently the areas must then

* There will be slight variations with the initial conditions, but these are not great and will not be discussed in this elementary treatment.

increase. Obviously there must be a neck where the converging and diverging portions of the nozzle join.

(e) It has been seen that, corresponding to each cross section of the nozzle, the steam has a definite condition and velocity, as shown by the ordinates in Fig. 456 immediately above the section under consideration. If the cross sections of the nozzle were shifted or spaced differently, so as to alter the longitudinal section of the nozzle from that shown, the ordinates of the curves would be similarly shifted and the character of the curves would change. This shifting may be so done as to cause any one of the curves to become a straight line; thus the nozzle may be such as to cause a uniform drop in pressure throughout its length, or a uniform increase in velocity, or a uniform increase in volume, whichever is most suitable for the purpose in hand, or any line can be made to have any desired curvature.

Usually nozzles are made with rounded entrance like that shown, but with straight conical divergence, as in Fig. 457, as this form is easiest to make and appears to be about as efficient as any. The stream lines of the jet issuing from such a nozzle are practically parallel, which is important if the jet is to act on turbine blades. For such a nozzle it is only necessary to compute the areas of the neck and discharge end; the entrance is then made rounded and the diverging portion is conical, the length depending on the angle of divergence, which should not be too great.

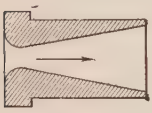


Fig. 457.

(f) With a given nozzle (ideal or approximately such) it is found that, accompanying a lowering of the pressure against which it discharges, the velocity of flow increases until that through the *smallest section* (or neck) reaches the critical value (the pressure then being the critical one) and that any further diminution of the pressure does not change the velocity and pressure at that section, nor does it increase the weight of material flowing per second through the nozzle. Hence, if the terminal pressure is below $.58\ p_1$, the area of the neck and the critical velocity will fix the amount of material flowing per second. If merely an orifice with rounded entrance is used the discharge velocity will be the critical regardless of the pressure against which the jet issues, provided it is below the critical. After the steam has once passed the neck it can further expand

in a properly proportioned nozzle and can acquire in the diverging portion of the nozzle an increased velocity of any desired amount (theoretically), its value merely being dependent on the terminal pressure.

332. Actual Steam Nozzles. (a) It will be remembered that adiabatic conditions are those under which the working substance, while undergoing a thermodynamic change, neither receives nor surrenders heat as such.

It has already been shown that in the actual case of flow through nozzles the conditions are practically adiabatic, for the time of contact of each particle with the wall is infinitesimal and, with continuous flow in one direction, each portion of the nozzle wall becomes heated to the temperature of the contiguous fluid and remains at that temperature; hence, neglecting radiation and conduction, there is no temperature head to produce heat transfer.

(b) But although the conditions are adiabatic the expansion process is not necessarily the equivalent of an isentropic one. In fact, it is possible to obtain adiabatic conditions of expansion, from the higher pressure to the lower one, under which none of the potential energy may be converted into kinetic energy of a flowing stream. For example, if the expansion is through a porous plug with pressure drop, the velocity of flow is negligible and, as has already been seen, the total heat ΔQ_2 per pound of working substance at the end of the process is equal to the amount ΔQ_1 it had at the beginning, hence $\Delta Q_1 - \Delta Q_2 = 0$ and $\Delta K = 0$. This expansion from the higher to lower pressure is along the constant heat lines on the Tϕ-diagram and on the Mollier Chart and is accompanied by increase in entropy. This is a case of resisted flow, and it can be considered that, for each slight pressure drop, some of the intrinsic energy $(\delta S + \delta I)$ of the material is expended in producing velocity (δv) of flow through a short length of plug, but that the friction and eddy currents reconvert this energy (δK) of flow back into heat (δQ) which is returned to the fluid and brings its stock back to the original value. Throttling of steam is a similar process.

(c) Between the constant heat expansion (with $\Delta K = 0$) and the isentropic one (with $\Delta K = \Delta E$) there may be an unlimited number of processes even though under adiabatic con-

ditions. It is, of course, desirable to so proportion the nozzle that it will offer no resistance to expansion, and cause no eddying, also that it will deliver the material in parallel stream lines. If this is effected the ideal conditions exist and the expansion is equivalent to isentropic.

(d) With resisted flow the velocity of the steam can be computed by using Eq. (505) and introducing the efficiency coefficient *Ef*, the proper values of which depend on the character, extent and shape of the guiding walls, and on the velocity, density and quality, or superheat, of the steam. The values of *Ef* range from .85 to .97 in nozzles used in turbines.

(e) The case of resisted flow is shown on the *Tφ-diagram* in Fig. 458, the heat ΔQ_1 initially associated with each pound of working substance being shown by the area surrounded by the heavy line. If the expansion were ideal, from 1 to 2, the heat ΔQ_2, at temperature T_2, remaining in the material after the process would be shown by the area below *ab*2. However, with resisted flow to the same lower temperature, T_2, less heat than the ideal amount is abstracted; hence more heat (of total amount $\Delta Q_2'$) remains associated with the material, its amount being shown by the sectioned area. The final state point is then at 2′; the quality is x_2', which is higher than x_2 as would be expected, and ϕ_2 is the entropy, which is greater than ϕ_1.

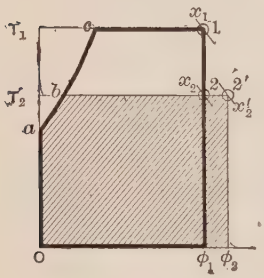


Fig. 458.

Evidently the heat available for accelerating the jet is $\Delta K' = (\Delta Q_1 - \Delta Q_2')$ which may be used under the radicals in Eqs. (504) and (505) to obtain the velocity of flow for this case; after which Eqs. (507) and (508) may be used in the same manner as before. If *Ef* is the efficiency of conversion then $\Delta K' = \Delta K \times Ef$, where ΔK is the energy theoretically available in the ideal case. In the Tφ-diagram there is no one area representing this available energy $\Delta K'$; it is merely shown by the difference between the area surrounded by the heavy line and that which is hatched.

(f) The *Mollier diagram* for resisted flow is shown in Fig. 459. With ideal expansion from state point 1 to 2, the heat per pound of steam would change from ΔQ_1 to ΔQ_2, the heat

surrendered would be $\Delta K = \Delta Q_1 - \Delta Q_2$, and the final quality would be x_2. With resisted expansion to the same terminal pressure, p_2, more heat, $\Delta Q_2'$ ($> \Delta Q_2$), remains in the steam than in the ideal case (as less is abstracted) and the final state point is found at the intersection 2′ of the constant heat line of $\Delta Q_2'$ with the pressure line p_2. The final quality x_2' ($> x_2$) and entropy ϕ_2 ($> \phi_1$) are determined by the quality and entropy lines passing through point 2′.

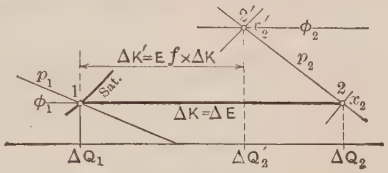


Fig. 459.

The heat converted into kinetic energy is $\Delta K' = \Delta Q_1 - \Delta Q_2'$ ($< \Delta K$), and as before $\Delta K' = \Delta K \times Ef$. This value of $\Delta K'$ is then used in the manner described at the end of the preceding paragraph.*

(g) The values of $\Delta K'$ and x_2' may also be found by computation. First ΔK for the ideal case is determined by the methods given in Sect. 94 for the Clausius cycle. Then with Ef assumed, $\Delta K'$ is $\Delta K \times Ef$, and the heat remaining per pound of steam at the end of the process is

$$\Delta Q_2' = \Delta Q_1 - (\Delta K \times Ef). \quad . \quad . \quad . \quad . \quad (509)$$

Since $\Delta Q_2' = x_2' r_2 + q_2$, it follows that

$$x_2' = (\Delta Q_2' - q_2) \div r_2, \quad . \quad . \quad . \quad . \quad . \quad (510)$$

where q_2 and r_2 correspond to the known terminal pressure.

The final entropy is then

$$\phi_2 = \phi_{l_2} + \frac{x_2' r_2}{T_2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (511)$$

in which all quantities on the right-hand side of the equation are either known or can be obtained from the steam table.

(h) The purpose of the foregoing discussion is merely to present the fundamental thermodynamic theory underlying the flow of steam at high velocities through nozzles and orifices. A detailed discussion of the subject cannot be attempted in this book and for further study the student is referred to any of the numerous textbooks on Steam Turbines.

333. Empirical Formulas for the Flow of Steam through Orifices. (a) It is sometimes convenient to have simple empirical formulas for quickly determining the approximate velocity of

* The Ellenwood Chart can be used similarly.

discharge from a given orifice, or for obtaining the area of orifice required for discharging a given weight of steam per second, when the discharge pressure is below .58 p_1. The following two formulas apply to such cases when the orifice has a properly rounded entrance, and they also apply to the neck of a diverging nozzle.

(**b**) *Napier's* experimentally determined rule gives the pounds of steam, initially dry, flowing per second from an orifice to be

$$w = (p \times a) \div 70, \quad . \quad . \quad . \quad . \quad . \quad (512)$$

or the area, in square inches, is

$$a = 70\, w \div p, \quad . \quad . \quad . \quad . \quad . \quad . \quad (513)$$

where p is the absolute pressure in pounds per square inch.

(**c**) A slightly more accurate but less convenient formula is that due to *Grashof*. For steam initially dry it is

$$w = \frac{p^{.97} a}{60}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (514)$$

from which

$$a = \frac{60\, w}{p^{.97}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (515)$$

the notation being the same as in (**b**).

334. Flow of Steam through Pipes. (**a**) The law for the frictional resistance accompanying the flow of steam through pipes resembles closely that expressed in Eq. (409) which shows that the resistance is directly proportional to the length of pipe, perimeter of cross section, character of surface and square of the velocity, and inversely to the area of section. To overcome this resistance and the inertia of the fluid there must be a drop in pressure from the entrance to the pipe to the discharge end. As the resistance varies with the square of the velocity, the pressure drop increases very rapidly as the velocity is made greater. Hence in practice the velocities used in pipes are very low, as compared with those in nozzles where the short length makes the loss by resistance very small even though great velocities are used.

(**b**) While the same treatment that is used for nozzles might be applied to pipes, the pressure drops and energy expended in producing the flow are so small that it is convenient to use

another method which disregards altogether the quantity of energy expended and only considers the pressure drop that is involved. This method makes use of Eq. (410), which for round pipes becomes

$$\Delta P = \frac{4}{2g} f\delta \frac{Lv^2}{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (516)$$

If W pounds of steam flow per minute, the volume flowing per second is the product of the weight per second ($W/60$) and the specific volume V, which is the reciprocal of the density δ. If the diameter of the pipe in inches is d, the velocity of flow is obviously $v = (144\ W\mathrm{V}) \div (60\ \pi d^2/4)$. Substituting this in Eq. (516) and solving gives the pipe diameter in inches,

$$d = c\sqrt[5]{\frac{W^2 L\mathrm{V}}{\Delta P}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (517)$$

where

W = pounds of steam flowing per minute,
L = length of pipe in feet,
V = specific volume of the steam,
ΔP = pressure drop throughout the length of pipe, and
c = a constant whose value is ordinarily .2, but this may be decreased slightly with very large pipes, as f seems to diminish somewhat as the diameter is increased.

The longer the pipe is made and the smaller the pressure drop allowed, the larger will be the diameter. But a larger diameter means increased first cost and greater heat loss by "radiation." Hence the diameter selected should be a compromise based on all these considerations.

(c) If the allowable velocity (v_m) of flow in feet per minute and the total volume (V_m) of steam to be transmitted in the same length of time are known, then the area of pipe in square inches immediately follows from

$$a = \frac{V_m \times 144}{v_m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (518)$$

For steady flow in high-pressure steam mains v_m is generally about 6000 feet per minute for saturated steam, while with superheated steam, with the larger sizes of pipe, and with those of short length, somewhat higher values prevail.

Exhaust pipes from turbines to condensers have velocities as great as 24,000 feet per minute and even higher in some cases.

(d) Steam engines receive and exhaust the steam intermittently and the area of pipes is commonly obtained by using Eq. (274) with $v = 6000$ to 7000 feet per minute for high-pressure live steam pipes and $v = 3500$ to 5000 for exhaust pipes.

335. Application of Steam Nozzles. The largest field of application for steam nozzles is in steam turbines, which have already been considered. Another wide field is in *Steam Injectors*, used for delivering feed water to boilers, and for similar purposes. This piece of apparatus, in its simplest form, is shown diagrammatically in Fig. 460. Briefly it operates as

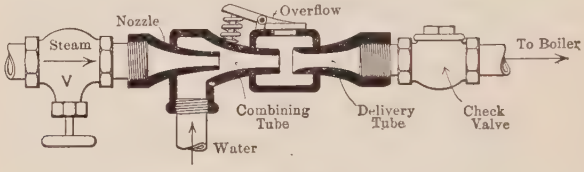


Fig. 460.

follows: Steam, admitted through valve V, acquires high velocity in passing through the nozzle, is condensed by the water in the combining tube and drives the water through the delivery tube and check valve into the pipe leading to the boiler. Thus, the flow through the nozzle is similar to that in the ordinary case, and the kinetic energy of the jet is used for injecting the water into the boiler against the pressure existing there.

Steam nozzles are also used for inducing draft in the stacks of locomotives and traction engines, the exhaust steam being used for the purpose, which results in a slight increase in the back pressure on the engine.

336. Perfect Flow of Ideal Gas. (a) In order to apply the equations of Sect. 329 to the flow of gases, it is first necessary to determine the intrinsic energy, $\Delta S\ (+\ \Delta I = 0)$, per pound of material. It was shown in Sect. 30 that during isentropic expansion the work performed per pound of gas is, from Eq. (50a),

$$778\,\Delta E = \frac{P_1\mathbf{V}_1 - P_2\mathbf{V}_2}{\gamma - 1} = 778\,\Delta S,$$

which is done at the expense of the intrinsic energy. If it is assumed that this same law prevails with expansion continued

to $P_2 = 0$, then all the intrinsic energy is converted into external work. Thus, upon this assumption, the total intrinsic energy per pound of material is, in general,

$$\Delta S = \frac{P\mathbf{V}}{778\,(\gamma - 1)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (519)$$

Then from Eq. (496) the potential energy per pound is (since $\Delta I = 0$),

$$Po = \left(\frac{P\mathbf{V}}{\gamma - 1} + P\mathbf{V}\right) \div 778$$

$$= \frac{\gamma}{(\gamma - 1)}\,(P\mathbf{V}) \div 778. \quad . \quad . \quad . \quad (520)$$

This is measured above a datum of absolute zero of pressures and temperatures, but as all problems in flow involve only differences in energies this fact need cause no inconvenience.

(b) Substituting in Eq. (501), for points 1 and 2 in Fig. 452, the values of Po_1 and Po_2 in terms of Eq. (520), gives

$$\left(\frac{v_2^2}{2\,g} - \frac{v_1^2}{2\,g}\right) = \frac{\gamma}{\gamma - 1}\,(P_1\mathbf{V}_1 - P_2\mathbf{V}_2). \quad . \quad . \quad . \quad (521)$$

If $v_1 = 0$, then the final velocity is

$$v = c\sqrt{2\,g\frac{\gamma}{\gamma - 1}\,(P_1\mathbf{V}_1 - P_2\mathbf{V}_2)}, \quad . \quad . \quad . \quad (522)$$

in which c is a discharge coefficient with value equal to unity in the ideal case. Substituting this value of v in Eq. (508), with $x = 1$, the weight discharged per second is

$$w = \frac{a}{144\,\mathbf{V}_2}\,c\sqrt{2\,g\frac{\gamma}{\gamma - 1}\,(P_1\mathbf{V}_1 - P_2\mathbf{V}_2)}. \quad . \quad . \quad (523)$$

All quantities on the right-hand side of this equation are generally known at the outset except $\mathbf{V}_2$. This latter must be determined before a solution can be effected. With isentropic expansion its value is found from the relation $P_2\mathbf{V}_2^{\gamma} = P_1\mathbf{V}_1^{\gamma}$ to be $\mathbf{V}_2 = \mathbf{V}_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}}$.

(c) Substituting the value of $\mathbf{V}_2$ in Eqs. (522) and (523), and simplifying gives

$$v = c\sqrt{2\,g\frac{\gamma}{\gamma - 1}\,P_1\mathbf{V}_1\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}\right\}}, \quad . \quad . \quad (524)$$

$$\text{and} \quad w = \frac{a \times c}{144\, \mathbf{V}_1} \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} \sqrt{2\, g \frac{\gamma}{\gamma - 1} P_1 \mathbf{V}_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} \right\}} \quad . \quad . \quad (525)$$

From which the area needed to discharge a given weight is

$$a = (144\, w\, \mathbf{V}_1) \div \left[c \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} \sqrt{2\, g \frac{\gamma}{\gamma - 1} P_1 \mathbf{V}_1 \left\{ 1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}} \right\}} \right] \quad (526)$$

As sections 1 and 2, in Fig. 452, may be located at any points along the conduit, or nozzle, it is possible to use these formulas to analyze the changes occuring between any two sections, or over the whole length of passage.

(d) If a curve is plotted to show how v, in Eq. (524), varies as (P_2/P_1) is decreased, it will be found that, as in the case of steam, the nozzle will have a neck if P_2 is low enough, and that the velocity through this neck becomes a maximum when a certain value of P_2/P_1 is reached. By differentiating Eq. (524) and making $dv/d(P_2/P_1) = 0$, this maximum velocity is found to occur when

$$P_2/P_1 = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}}, \quad . \quad . \quad . \quad . \quad . \quad (527)$$

the value of which is .527 when $\gamma = 1.41$. Thus in this case the maximum or *Critical Velocity* at the neck occurs when P_2 has been reduced to a critical pressure of .527 P_1.

This phenomenon has been repeatedly verified experimentally. It is found, as with steam, (1) that at the neck the critical pressure is, in the ideal case, always about .527 P_1, provided the pressure beyond the neck is equal to or less than this amount; (2) that lowering the discharge pressure below the critical pressure changes neither the pressure nor the velocity at the neck; and (3) that the neck velocity and initial PV-condition determine the maximum amount of working substance which can flow through a given orifice or nozzle. Equations (524) and (525) will therefore give the maximum ideal velocity and weight of discharge through a neck of given cross section if .527 is substituted for (P_2/P_1); and Eq. (526) can be used to compute the neck area required for discharging a given weight of gas, if similar substitution is made.

(e) In Sect. 87 (f) it was shown that when steam expands isentropically, for an initial quality greater than 70 per cent, the

process is represented quite accurately by the equation for the isentropic expansion of gas, with an exponent n equal to $(1.035 + 0.1\,x)$, in which x is the quality fraction. Thus it follows that Eqs. (524) to (526) are applicable to the flow of steam through orifices if this value of n is substituted for γ.

(f) When the quality is unity, $n = 1.135$, and if this value is introduced for γ in Eq. (527) it is found that the *maximum flow of dry saturated steam through an orifice occurs when* $P_2/P_1 = .58$ (about). This is the value found by the method given in Sect. 331 and its correctness has been verified experimentally.*

337. Imperfect Flow of Gases. (a) The actual velocity and weight of discharge from an orifice, or nozzle, are of course less than the theoretical. There are a number of reasons for this: — The real gases differ somewhat from the ideal; friction prevents some of the available heat energy from becoming converted into kinetic energy of flow; some of the kinetic energy is wasted in producing eddy currents; heat energy is lost by radiation and conduction; and, if an orifice has improperly shaped walls, the cross section of the jet at its neck may be less than that of the orifice, as indicated in Fig. 461. To make allowance for the contraction of area, and for the various losses, the *discharge coefficient* c is introduced in Eqs. (524) to (526). The value of this coefficient varies from .56 for certain sharp-edged circular orifices to nearly unity in the case of a mouth with properly rounded entrance.

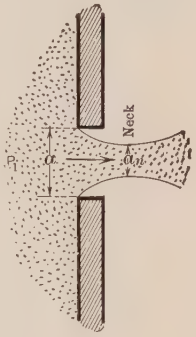


Fig. 461.

* For superheated steam γ is about 1.3 and $P_2/P_1 = .546$.

CHAPTER XLI.

COMPRESSED AIR.

338. Definitions. (a) Air compressors in the broadest sense are all devices used for raising the pressure of air, but technically the term is generally applied only to apparatus which raises the pressure to a comparatively high value, say some value between 25 and several hundred pounds per square inch. In extreme cases the pressure is increased to several thousand pounds per square inch.

(b) Other devices, such as fans and rotary blowers, are really compressors but are seldom spoken of as such, principally because the pressures attained are so small that the principal function may be considered to be the propelling of air rather than its compression.

(c) The term Blowing Engines, or Blowers, is used to designate certain apparatus used for compressing air to pressures between about 10 pounds and 30 pounds above atmospheric for use in blowing cupolas and blast furnaces. These are properly air compressors but because of the low pressures many of the difficulties attending compression to higher pressures are not met in their design.

339. Elementary Air Compressor. (a) The essential parts of an ideal air compressor of the simplest kind are shown semi-diagrammatically in Fig. 462, *A* being the spring-closed admission or inlet valve, which opens inwardly, and *B* the spring-closed discharge valve, opening outwardly. In the simplest case there will be no clearance, the piston just touching the cylinder head at one end of its stroke.

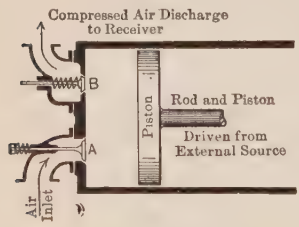


Fig. 462.

(b) Imagine the piston in contact with the cylinder head in the ideal case. By the application of an external force to the piston rod the piston can be drawn to the

right, and air will then enter the cylinder through valve A at atmospheric pressure P_1, according to the horizontal line ab, in Fig. 463.

(c) When the cylinder has been thus filled with air, the piston may be driven back to the left. As soon as such motion starts the valve A will be closed by the light spring shown, and the air entrapped in the cylinder will then be compressed according to some law, as shown by bc, the final volume being dc.

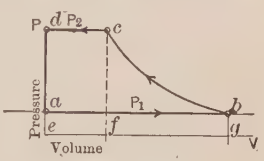


Fig. 463.

There are two limiting conditions which may be imagined as existing during compression:

(1) No heat may be removed from the air during the process, in which case the compression will be *adiabatic* (with rise in temperature); and

(2) All the heat generated during compression may be removed, in which case the compression will be *isothermal*.

All actual cases generally fall between these limits as will be seen later.

(d) Imagine the discharge pipe to be connected to a closed vessel, "a receiver," in which is maintained a constant pressure, P_2, equal to P_c. Assume further that the action of this pressure upon the valve B, plus the action of the spring is such that a uniform pressure of P_c pounds per unit area of valve face will just balance it.

Then when the piston has compressed the air to the pressure P_c the discharge valve will open (at c) and the continued motion of the piston will "discharge" the air at constant pressure, P_2, as shown by the line cd. Thus, the piston will end its stroke in contact with the cylinder head, having discharged at pressure P_2 all the air received at pressure P_1.

340. Work Done in Compressor. (a) In Fig. 463, area $abge$ shows the work done *upon* the piston during the outstroke by the entering air, $gbcde$ represents that done *by* the piston on the air during the instroke, and the net work is shown by area $abcd$.

Thus the area of the "compressor diagram," or card, measures the net work done *by* the piston upon the air, just as the area of an engine diagram measures that done *upon* the piston by the working substance.

The Compression Line.

(b) In Fig. 464 are given two superposed diagrams, *abcd* and *abc'd*, both from the same ideal compressor which is to compress to a pressure $P_2 = P_d$ the amount of air which originally occupied the volume V_b cubic feet when at atmospheric pressure. The compression line *bc* is an *isothermal* and the line *bc'* is an *adiabatic*.

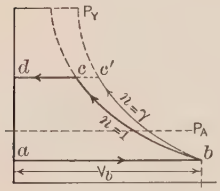


Fig. 464.

It is evident from the figure that the diagram containing an adiabatic compression line encloses more area than that having isothermal compression, and hence more energy from some outside source will be required per cycle. Obviously, the isothermal compression is the more desirable, other things being equal, the work saved over that with adiabatic compression being shown on the diagram by the area *cbc'*. The higher the compression pressure the greater is the ratio of the work done with adiabatic compression to that with isothermal.

(c) With isothermal compression the temperature of the air at *c* and *b* is, of course, the same, but during adiabatic compression the temperature rises according to Eq. (51), the final temperature being

$$T_c' = T_b\left(\frac{V_b}{V_c'}\right)^{\gamma-1} = T_b r^{\gamma-1} \quad . \quad . \quad . \quad . \quad (528)$$

in which r = ratio of compression $= \dfrac{V_b}{V_c'}$.

If the air with pressure P_c' and volume V_c' is cooled at constant pressure it will attain a volume V_c when the initial temperature T_b is reached. This is approximately what happens in most real cases, for after the cooling has occurred the air is in the same condition as though it had been compressed isothermally. It is therefore advisable to strive for isothermal compression if this can be attained, or approached, without entailing greater outlay than the cost of the work area *cbc'*.

Formulas for Work.

(d) From the diagrams of Fig. 463 and Fig. 464, the work done by the piston per cycle with *isothermal compression* and no clearance is evidently

$$\text{Work} = \text{work on } bc + \text{work on } cd - \text{work on } ab, \text{ ft.-lbs.}$$

$$= P_c V_c \log_e \frac{V_b}{V_c} + P_c V_c - P_b V_b, \text{ foot-pounds. . .(529)}$$

With *adiabatic compression* the work is (Fig. 464)

$$\text{Work} = \text{work on } bc' + \text{work on } c'd - \text{work on } ab$$

$$= \frac{P_c' V_c' - P_b V_b}{\gamma - 1} + P_c' V_c' - P_b V_b, \text{ foot-pounds. . (530)}$$

With *any compression curve* expressible by the equation PV^n = constant, the work per cycle is

$$\text{Work} = \text{work on } bc'' + \text{work on } c''d - \text{work on } ab$$

$$= \frac{P_c'' V_c'' - P_b V_b}{n-1} + P_c'' V_c'' - P_b V_b, \text{ foot-pounds. (531)}$$

341. The Effect of Clearance. (a) No real compressor can be operated with the zero clearance assumed for the preceding elementary consideration. There must always be a certain amount of mechanical clearance between cylinder head and piston to insure safe operation and there are always passages or ports of some sort between the valve faces and the inside of the cylinder.

(b) In Fig. 465 is given an ideal compressor diagram for a machine with clearance volume $V_{cl} = V_d$. At the end of the discharge, that is, after the completion of the constant-pressure process *cd*, the clearance contains $V_{cl} = V_d$ cubic feet of air at a pressure $P_2 = P_d$. When the piston starts on the outstroke the inlet valve will be held closed by this pressure until the piston has moved out far enough to allow the clearance air to expand to atmospheric pressure according to some such process as *da*. When *a* is reached the inlet valve will open and during the remainder of the stroke external air will enter the cylinder, as in the previous case. Though the stroke of the piston is such as to make available the volume $V_b - V_a' = V_s$, the amount of air actually entering the cylinder will be $V_b - V_a = V_{es}$ and only a fraction of the stroke, equal to V_{es}/V_s, has therefore

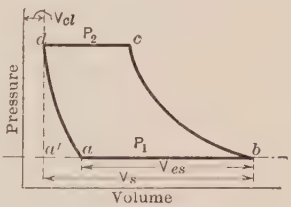


Fig. 465.

been usefully employed. Thus the *volumetric efficiency* of this ideal simple compressor must be

$$VEf = \frac{V_{es}}{V_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (532)$$

Obviously the piston displacement of the compressor with clearance must be to that of the compressor without clearance in the proportion V_s/V_{es} if both are to compress the same quantity of air per cycle, therefore the existence of clearance makes necessary a larger compressor to handle a given volume of air.

Effect of Clearance upon Work.

(c) In the ideal case it may be assumed that the expansion of the clearance air along *da* takes place according to the same law as the compression of the mixture of clearance air and cylinder charge along the curve *bc*. Then, if the clearance air be imagined to be separated by a flexible diaphragm from the fresh charge, and to be used over and over again, it is evident that it will deliver just as much work when expanding from *d* to *a* as will subsequently be consumed in compressing it from *a* to *d*. Then the net work necessary per cycle will be only that required to compress the volume V_{es} of cylinder charge from pressure P_1 to P_2 and discharge it at P_2. Therefore, in the ideal case the presence of clearance does not alter the net work which must be done per cycle.

It should, however, be noted that since the compressor with clearance will be larger than that without, the friction losses and cost of the machine will be greater in the real case.

342. **Real Single Stage Compressor Diagram.** (a) The real compressor differs from the ideal in many respects, chiefly because the cylinder and piston cannot be made of heat-resisting materials, because the valves cannot be made to operate perfectly, and because of the inertia of the air being handled.

A diagram obtained from a real compressor is shown in Fig. 466, superposed upon an ideal one for the same machine, the pressure of air being supposed to be raised from atmospheric (P_1) to a receiver pressure equal to P_2.

(b) The ideal card is drawn for isothermal compression of the clearance air and charge and with isothermal expansion of the

clearance air. In the real case it is never possible to obtain isothermal compression with a reciprocating air compressor; instead, the compression line falls between the adiabatic and the isothermal and is expressible by the equation PV^n = constant, with values of n varying from about 1.2 in extremely favorable cases to about 1.3 under rather unfavorable conditions. To obtain such a curve it is necessary to cool the air during compression, by the methods which will be considered later. For present purposes it is merely necessary to note that the air, and therefore the cylinder walls, will become heated during compression.

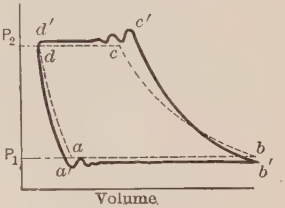


Fig. 466.

(c) During expansion of the clearance air, this material will, in general, be in contact with walls which are at a higher temperature, hence it will receive heat during the process. Ordinarily the real expansion line for this air lies between an adiabatic and an isothermal, but in the average case it approaches more nearly to the latter. The real expansion line may then be assumed to have a shape and position similar to da' in the figure.

(d) In the ideal case the admission valve would open as soon as the clearance pressure has decreased to atmospheric, but actually the pressure must drop somewhat lower to give an unbalanced pressure great enough to open the valve against its spring, its friction and inertia, and also to overcome the inertia of the air and the resistance to flow through the more or less restricted areas available.

After the valve is open and the air is in motion there are generally several oscillations of the valve and the air column, as indicated by the wavy suction line, after which the pressure settles down to an average value sufficiently below the atmospheric pressure to cause the inflow. The oscillation of the valve is known as "fluttering."

(e) From the fact that the same machine has been assumed in both cases it is evident that the actual *volume* of air in the cylinder at the end of the suction stroke must be the same in each case. In the real case, however, the air has a lower pressure than it has in the ideal, and, in general, its temperature will also have been raised by the heated walls which are uncovered

by the piston, thus the actual *weight* present will be less than the ideal. This effect will be considered more in detail in a later section.

(f) Starting at the point b', instead of b, the air in the real case will be compressed according to some law intermediate between the adiabatic and isothermal and therefore steeper than the ideal, as shown by $b'c'$ in the figure.

(g) The discharge valve does not open until the pressure attained is slightly above that in the receiver and it behaves much like the suction valve and for similar reasons. The discharge line actually obtained will then generally look something like $c'd'$ instead of cd.

(h) The discharge valve will obviously not close suddenly at the end of the stroke, consequently the corner at d' may be more or less rounded, the exact point at which expansion of the clearance air starts being rather difficult to determine.

(i) The net result of the operation of the real compressor has been to compress a smaller weight of air than that handled by the ideal machine and to require the expenditure of work in excess of the ideal, as shown by the greater area enclosed.

343. Volumetric Efficiency. (a) The volumetric efficiency of the ideal compressor was shown to be V_{cs}/V_s, the symbols referring to Fig. 465. In a real case the determination of the volumetric efficiency is not as simple as this and it is often very difficult, if not impossible, to obtain its true value. As a result several incorrect volumetric efficiencies easily obtainable from a card are often used in practice.

True Volumetric Efficiency.

(b) An ideal compressor should receive a charge equal to its total piston displacement and this charge should have atmospheric temperature and pressure. The weight of this charge would then be

$$W = \frac{V_s}{V_a},$$

in which V_s = piston displacement in cubic feet, and

V_a = volume occupied by one pound of air under atmospheric conditions.

If, in any case, the weight of air actually received per suction stroke is W', the true volumetric efficiency is

$$VEf_t = W'/W \quad . \quad . \quad . \quad . \quad . \quad (533)$$

$$= \frac{V_a W'}{V_s}. \quad . \quad . \quad . \quad . \quad . \quad (533a)$$

(c) To evaluate this efficiency it is necessary to determine W' and this is generally very difficult to accomplish in any real case. It can be done by measuring the air actually received or discharged in a given time and then dividing this by the number of suction strokes occurring during that time period; but the accurate measurement of large quantities of air is generally difficult and therefore the true volumetric efficiency is seldom determined.

Atmospheric Volumetric Efficiency.

(d) In Fig. 467 is given in exaggerated form a real compressor diagram with the atmospheric line added. If there is no change of temperature of the working substance during charging and if the expansion of the clearance air and the compression of the mixture both follow the same law, the distance *ef* must measure the volume occupied by the clearance air, and *fg* is that occupied by the charge when compression has progressed up to the point *g*.

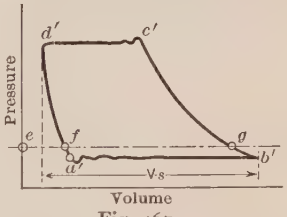


Fig. 467.

If it be further assumed that the drop of temperature from *f* to *a'* and the rise from *b'* to *g* are negligible, it may be said that the distance *fg* is a measure of the volume occupied by the charge when at atmospheric pressure and temperature. Then the volumetric efficiency, on an atmospheric line basis, would be

$$VEf_a = \frac{\text{distance } fg}{V_s}. \quad . \quad . \quad . \quad . \quad . \quad (534)$$

(e) This formula will probably give an incorrect result in all cases, because none of the assumptions made in its derivation are strictly correct. However, the error is generally not great in magnitude, hence, because of the simplicity of the method, the formula is commonly used in practice.

Suction Line Volumetric Efficiency.

(f) A still less perfect formula is often used for determining the volumetric efficiency from the diagram. The point a', Fig. 467, is first determined by continuing the straight part of the suction line backward to intersect the expansion line. The distance $a'b'$ is then taken as the volume occupied by the charge and the distance from a' to the zero of volumes is taken as that of the clearance air at the same pressure and temperature. Neglecting the fact that the suction line is below atmospheric pressure the volumetric efficiency on a suction line basis is defined as

$$VEf_s = \frac{\text{distance } a'b'}{V_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad (535)$$

(g) This method contains practically all the errors of that previously given with the added disadvantage of neglecting the difference of pressure, consequently it should never be used when the position of the atmospheric line can be obtained.

344. Cooling During Compression. (a) It has been shown in preceding sections that isothermal compression should, in general, prove more economical than adiabatic. It is practically impossible to attain isothermal compression in any real machine, but it can be more or less closely approximated and generally without involving excessive cost.

(b) If a compressor fitted with a metal cylinder is operated very slowly, i.e., one or two cycles per minute, the heat generated by compression will be conducted away by the cylinder metal almost as fast as generated and in such case the compression could be made to approach an isothermal process as closely as desired, but it would involve the use of enormously large machines because of the slow operation. Some method must therefore be used which permits of operation at the highest desirable speeds.

(c) If any form of external cooling, such as radiation considered above, is to be used the proportions of the cylinder are important. That cylinder which exposes the greatest surface to the external cooling agent, per cubic foot enclosed, will be the best so far as cooling is concerned. This would indicate the use of cylinders of small diameter and great length, i.e., "long stroke compressors," but such machines are always more

expensive than short stroke mechanisms, consequently a commercial limit is set to cylinder proportions adopted.

(d) In practice a few types of compressors are cooled by radiation to the atmosphere, as, for instance, those used on locomotives for operating the air brakes. They are all comparatively small, are generally operated in strong currents of air and are at best rather inefficient. It is doubtful if the compression is appreciably better than adiabatic, the radiation serving simply to prevent overheating of the entire mechanism by storage of heat from cycle to cycle.

(e) Most commercial machines are water cooled. There are three distinct methods of applying cooling water, two or more of which may be, and generally are, used on the same machine. They are: — (1) Injecting water into the cylinders; (2) water jacketing the cylinders; and (3) compressing in stages and using water jacketed vessels, called "intercoolers," between cylinders.

Water Injection.

(f) The injection of water into the compression cylinder has been rather extensively used in Europe but not in this country. If the water is introduced as a solid stream but little cooling is effected, the compression curve approximating the equation $PV^{1.35}$ to $PV^{1.37}$ = constant; but with a very fine spray the cooling effect is much greater, and values of the exponent n as low as 1.26 to 1.28 may be obtained.

(g) The introduction of water into the cylinder has the following disadvantages: — It generally increases the wear of cylinder and piston; the feeding devices are an almost constant source of trouble; and the air leaves the cylinder practically saturated with water, some of which precipitates when cooled in the receiver, but much remains in the air and later may cause trouble by freezing when the air is expanded in doing work.

Water Jacketing.

(h) Jacketing the compressor cylinders with water does not introduce the difficulties considered above, but it is generally less efficient than water spraying unless it is very perfectly carried out. Values of the exponent n about equal to 1.25 to 1.28 are generally attainable.

Multistage Compression and Intercooling.

(i) The raising of pressure from atmospheric to the desired receiver pressure need not occur entirely in one cylinder. The compression may be divided between as many cylinders as desired without changing the ideal process in any way. This is shown in Fig. 468 for the ideal case with three cylinders, i.e., compression in three stages. It can be seen from the diagram that it is immaterial whether:—

(1) Compression is carried out in one cylinder receiving the charge V_b and compressing it isothermally and discharging at a pressure P_2; or

(2) It is carried out in several cylinders, the first receiving a charge V_b at pressure P_1, compressing isothermally to f and

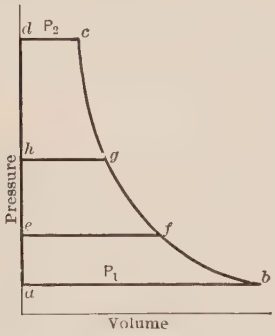


Fig. 468.

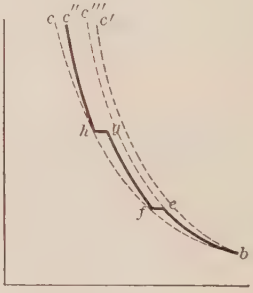


Fig. 469.

then discharging along *fe* to a second cylinder which, receiving its charge along *ef*, compresses to *g*, and so on until the last cylinder compresses to and discharges at P_2.

(j) This method has the practical advantage of making it possible to use what is known as "*intercooling*"; for the air discharged from one cylinder may be passed through a very efficient cooler on its way to the second, and so on. The practical advantage of this is shown in Fig. 469, which indicates the compression line which is thus made possible.

The line *bc* represents ideal or isothermal compression, *bc'* shows an adiabatic, and the broken line *befghc''* the compression line which might be obtained with a jacketed, or spray cooled, multistage compressor fitted with intercoolers.

The compression in the first cylinder brings the material to *e*

with a temperature higher than it had at *b*. An effective intercooler through which the air passes on its way to the next cylinder can reduce its temperature to the original value, so that compression in the second cylinder starts under the same conditions as though the process in the first had been isothermal.

The work done in excess of that required in the ideal case is evidently measured by the small areas *chc''*, *hfg* and *fbe*, while without intercooling the loss would probably have been something like that shown by the area *cbc'''*.

(k) The fact that the cooling water is often below the average atmospheric temperature suggests the possibility of cooling in the intercoolers to a value lower than that on the isothermal. This would give a compression line similar to that shown in Fig. 470 by *befghc''*, which is sometimes approximated in practice when very cold water is available. Comparing this with the isothermal *bc* shows that the latter may under these circumstances be very closely approximated. Some few machines have been operated with such effective intercooling that the sum of the work areas under the real compression lines in the several cylinders was less than under the ideal isothermal between the initial and final pressures.

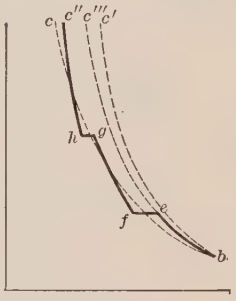


Fig. 470.

(l) Dividing the compression up into several stages and intercooling has a markedly beneficial effect upon the volumetric efficiency of a compressor for two reasons: —

First, since the temperature range in the low-pressure cylinder (1st stage) is reduced, the air is heated less during the suction stroke, hence greater actual weight will enter the cylinder than would be the case in a single cylinder operating between the extreme pressure limits.

Second, less *weight* of air remains in the clearance of the low-pressure cylinder because the discharge pressure from that cylinder is lower than with a single-stage compressor, which would be of the same size and have the same clearance volume.

This will be made clear by Fig. 471 in which the idealized cards of a three-stage compressor with clearance are *abcd*, *a'b'c'd'* and *a''b''c''d''*. It will be observed that the expansion of the

clearance air in the low-pressure cylinder theoretically decreases the charge volume by the small amount equal to $V_a - V_d$; whereas if the compression had all been carried out in this one cylinder the clearance air at pressure P_2 would have expanded from a volume V_e to a volume V_f, theoretically decreasing the charge by the very large amount $V_f - V_d$. This serious loss is one worth preventing, if commercially feasible.

(**m**) It is obvious from the diagrams and preceding paragraphs that the larger the total pressure range, the greater in every way will be the advantages of multistage compression. It thus happens in practice that machines for compressing to 25 or 50 pounds per square inch are generally built single stage, while

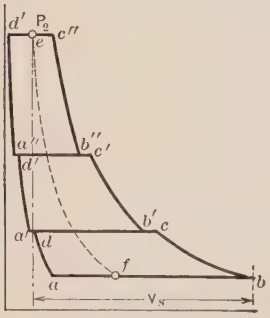


Fig. 471.

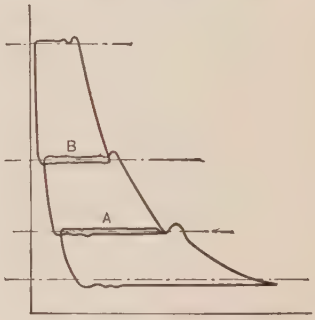


Fig. 472

those intended to compress to 100 or 150 pounds are generally made two-stage. Where exceptional efficiency is desired, or where extremely high pressures are to be attained, three-, and even four-, stage machines are sometimes used.

(**n**) The differences between the actual cards and the ideal ones of each cylinder of a multistage compressor are similar to those which have been discussed for the single-stage compressor. When superposed they look something like Fig. 472, in which the atmospheric, intercooler, and receiver pressures are indicated by horizontal dash lines. In each case the air is drawn into a cylinder at a pressure below that at which it exists outside of the cylinder and is discharged at a pressure higher than that maintained in the vessel receiving the air. This results in an overlapping of the cards in the center of the diagram giving two loops, A and B, which very evidently represent lost work.

(o) The better the action of the valves and the larger the passages through ports, intercooler, pipes and such, the smaller will the areas of these loops become, the upper and lower lines tending to become coincident. In very well-designed compressors this lost work is so small as to be almost if not quite indeterminate.

345. Blowing Engines. Blowing engines, or blowers, are similar to air compressors in principle, but they are generally built to handle relatively very large quantities of air at comparatively low pressures, say 10 to 20 pounds per square inch above atmosphere. Comparatively little attention need be given to cooling under such conditions because the pressure is so low that very little work can be saved by such means. Moreover, the compressed air is generally heated before being used, so that any cooling during compression would call for an expenditure of heat to raise the temperature immediately afterward.

Because of the large volumes of air to be handled considerable difficulty is generally met in designing efficient valves, particularly if operated at high speeds. As a result there are many different types of both inlet and discharge valves in use, some operating automatically under the action of springs and air pressure, some mechanically operated, and some partly automatically and partly mechanically operated.

346. Turbine Compressors. Since the successful commercialization of the steam turbine, engineers have been trying to develop satisfactory "Turbo Compressors" or "Turbo Blowers." These compressors have a number of stages arranged in series, each impeller receiving its supply of air from the preceding stage and discharging into the one which follows, no valves being used. The stages are water cooled and intercoolers are employed. These machines are just beginning to assume prominence for compressing to pressures from 10 to 20 pounds or more per square inch, but as yet few have been used in this country.

347. Compressed-air Engines. (a) Compressed air is used commercially in many different ways but most widely in engines for the production of power, the air serving as the working substance. At first sight it seems an uneconomical method of producing power as the air compressor must be driven by an

engine of some sort which apparently might better be used directly to produce the power desired, rather than to suffer the additional losses incurred during compression and utilization of the compressed air in a second engine.

(b) Such reasoning is generally sound for conditions where the desired power can be conveniently generated at the point of utilization by any of the prime movers previously considered. There are, however, many cases where this cannot be done. Where a number of small engines are to be operated at widely scattered points and where electrical transmission is not suitable, compressed air engines find a field to which they are admirably suited. Compressed air can be transmitted for great distances without appreciable loss, and, as will be shown later, any loss can be more than made good at the point of consumption. Steam, on the other hand, cannot be efficiently transmitted over great distance because of the resulting condensation; and more than this, steam engines of small size are very inefficient and the high temperature at which they operate renders them unsuitable when handling is required. The working substance of internal combustion engines can be transmitted as easily, if not more easily, than compressed air, but the complicated valve and ignition mechanisms, the high temperature and the hot noxious exhaust gases make them less desirable than compressed air engines for a number of purposes.

(c) Thus compressed air engines are widely used in mining and quarrying operations and for the driving of small portable tools in shops and such. In Paris there is installed a central compressor station which distributes compressed air, much as gas is distributed in this country, and air is used by the consumers for operating small plants much as electricity is used here.

348. Compressed Air Engine Cycles. (a) Compressed air is sometimes used in engines without expansion, that is, according to the rectangular cycle. The work done per cycle in an engine without clearance is obviously

$$\text{work} = V_b (P_2 - P_1), \quad . \quad . \quad . \quad . \quad . \quad (536)$$

in which V_b is the volume displaced per stroke, P_2 is the upper pressure, and P_1 that of discharge.

Such use of compressed air is very uneconomical as no use is made of its associated heat and, as a result, better methods of utilization have been devised.

(b) A cycle similar to the Clausius described under vapor cycles is generally considered to represent the ideal cycle for air. As shown in Fig. 463, it consists of two constant pressure lines, *dc* and *ba*, a constant volume line *ad*, and an adiabatic expansion line *cb*. The theoretical work made available by such a cycle can easily be determined from the formulas previously given.

(c) In practice the air generally enters an engine at about atmospheric temperature and during the approximately adiabatic expansion it becomes cooled, in some cases to such an extent that the moisture in it freezes and leads to difficulties. The theoretical temperature decrease can never be attained in any real engine because heat will be supplied to the engine cylinder from the surrounding atmosphere and will tend to make the expansion more nearly isothermal. If the engine were operated very slowly the expansion would very closely approach a true isothermal.

(d) This approach toward isothermal expansion is advantageous for the following reasons:

(1) It tends to prevent the deposition, in the form of ice, of the moisture accompanying the air, thus tending to prevent the resultant troubles with lubrication and stoppage of valves and passages.

(2) It increases the work made available, as the area under an isothermal is greater than that under the steeper adiabatic between the same two pressures.

(3) It lessens the range of temperature within the cylinder so that there is less tendency to cool down the entering air. Such cooling would result in a decrease of volume and therefore an increase in the weight required per cycle.

(e) It is interesting to note that from the theoretical viewpoint isothermal operation is not as advantageous as adiabatic. The object of using expansion is to make use of some of the heat associated with the working substance as it enters the engine. If the expansion is isothermal no work can be done at the expense of such associated heat; on the contrary, heat equivalent in quantity to the work done must be supplied from an external source. With an adiabatic expansion, however, all work would

be done at the expense of heat already associated with the gas as it enters the cylinder.

The discrepancy between theory and practice is due to the fact that in the assumed case heat supplied from the atmosphere during the isothermal expansion costs nothing and may therefore be freely used without decreasing the commercial efficiency of the process.

(f) It was shown that it was uneconomical to use complete expansion in a reciprocating steam engine. The same thing is true in the case of a reciprocating compressed air engine, and as a result the toe of the card is cut off in practice.

(g) Real engines are further found to operate more quietly, and therefore more satisfactorily, when the exhaust valve closes before the end of the stroke trapping some air which is then compressed into the clearance. Such operation causes a loss of diagram area and therefore a loss of work from a given size of cylinder running at a given speed. It may, however, result in a saving in the amount of air used per horse power in two ways:

(1) When compression is not used the air admitted must first be mixed with that in the clearance until the full admission pressure is attained; after that the entering air becomes available for driving out the piston, and

(2) Compression tends to raise the temperature of the walls, cylinder head and piston and thus to decrease the cooling effect upon the incoming air.

349. Preheating. (a) In practical use, compressed air engines and the compressors supplying the working fluid are generally widely separated. It has already been shown that so far as the compressor is concerned the cooler the air the better. A cool supply means larger capacity for a given machine and efficient cooling during operation means a smaller amount of work required. The same thing is true for the "receiver" or storage tank, and for the pipe line carrying the air to the engine, for the cooler the air the smaller can these parts be for a given quantity of air.

(b) Conditions are, however, quite different so far as the engine is concerned. The warmer the air, within reason, the better.

If the compressed air could be heated at constant pressure

before entering the engine, it would expand according to Charles' law. A given volume of heated air admitted to the engine would represent a smaller actual weight but would be able to deliver the same amount of work as a larger weight of colder air, and there would be the added advantage that there would be less danger of the moisture freezing at the end of expansion.

(c) Such heating of the air is known as "preheating" and the devices in which it is effected are called "preheaters." It is actually used in places where the transmission piping is of great length and also where the engine units are few and of large size. It is found in practice that the running expense for the fuel supplied for preheating is less than the extra charges against the larger compressor and pipe line which would otherwise be used.

CHAPTER XLII.

REFRIGERATION.

350. Definition. (a) By *refrigeration* is generally meant the removal of heat from a body, or substance, to such an extent as to leave it, or maintain it, at a lower temperature than that of its surroundings. This may be done commercially in moderate climates by the use of ice; it may be accomplished in the laboratory by the use of liquefied gases; it may be done in very hot climates by the naturally rapid evaporation of water.

(b) In the ordinary engineering application of the term, however, it is taken to mean the removal of heat by mechanisms, or systems, which will be described in later sections and which are grouped under the title of *Mechanical Refrigeration*.

351. Thermodynamics of Refrigeration. (a) It was shown in Sect. 49 (h) that by the expenditure of energy (ΔE) a reversed heat engine would remove heat from a body at low temperature and would discharge to another body, at higher temperature, that heat plus the heat equivalent to the energy expended in the operation. That is, a reversed heat engine shown diagrammatically as R in Fig. 473 can receive a stream of heat ΔQ_2 from the low temperature body T_2 and discharge the larger stream ΔQ_1, made up of ΔQ_2 and ΔE, to the high temperature body T_1.

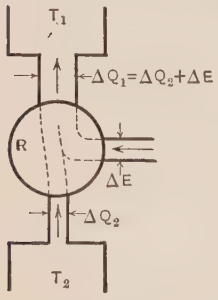


Fig. 473.

(b) This is a process of refrigeration because heat can be removed from the low temperature body even if its temperature be far below that of its surroundings. It thus appears that the reversed heat engine, which has been called a heat pump, is what may now be called a refrigerator, or refrigerating machine.

(c) Imagine the Carnot cycle, shown in Figs. 18 and 21 to PV- and Tϕ-coördinates, to be carried through in the direction

dcba for purposes of refrigeration. Heat will be absorbed along the line *dc* at temperature T_2, and in quantity as shown on the Tϕ-diagram by the area *dcef*. The work expended in driving the machine will be shown by the area *abcd* on the PV-diagram, if measured in foot-pounds, or by the similarly lettered area on the Tϕ-diagram, if measured in thermal units. Heat will be discharged along the line *ba* at temperature T_1 and its quantity will be the area *feba* on the Tϕ-diagram, equal to the sum of two areas previously considered.

(d) The expenditure made in order to abstract the heat ΔQ_2, shown in Fig. 21 by area *dcef*, is obviously the energy ΔE used in driving the machine as shown by the area *abcd*. If the heat removed, ΔQ_2, be taken as the result obtained, the efficiency of the process is

$$Ef = \frac{\text{Result}}{\text{Expenditure}} = \frac{\Delta Q_2}{\Delta E}.$$

(e) It is obvious from the Tϕ-diagram of Fig. 21 that for the case for which this figure was drawn ΔQ_2 is considerably greater than ΔE, hence the ratio which has just been given as the efficiency will be greater than unity. This is a very common property of refrigerating processes. Since engineers are not accustomed to speak or think of efficiencies greater than unity it is common practice to call this ratio the *Coefficient of Performance* (C.o.P.), or the *Figure of Merit*, rather than the efficiency of the process. Then the

$$\text{C.o.P.} = \frac{\Delta Q_2}{\Delta E} \quad . \quad . \quad . \quad . \quad . \quad . \quad (537)$$

for refrigerating machinery of this kind.

(f) The apparently remarkable attainment of an efficiency greater than unity is meaningless. The work expended and the heat removed from the cold body are really not connected in any such way as are heat supplied and work done in the case of an engine. This can best be seen from the Tϕ-diagram of Fig. 21. Assume the line *dc* to be moved upward while the line *ab* maintains its position. Then the heat removed (ΔQ_2) will obviously increase while the area of the cycle, representing ΔE, will decrease, that is, it takes less work to remove larger amounts of heat.

(g) That this should be so can easily be seen by carrying the

assumptions to the limit. If *dc* rises above *ab* the previously cold body has attained a temperature greater than the previously hotter one and work can actually be obtained by allowing heat to flow from it to the latter. Obviously the coincidence of the lines *dc* and *ab* would indicate that the two bodies are at the same temperature, that no work is attainable by heat flow, and that no work is necessary to cause heat flow.

Requirements for Maximum Coefficient of Performance.

(h) Inspection of the Tϕ-diagram will show that anything which brings the two lines *ab* and *dc*, that is, the temperatures T_1 and T_2, closer together will increase the value of the coefficient of performance. This can be done by dropping T_1 or by raising T_2. Dropping T_1 will decrease ΔE but will not change ΔQ_2. Raising T_2 will decrease ΔE and increase ΔQ_2 by the same amount. It is therefore evident that raising the lower temperature is more effective for attaining a high coefficient than lowering the upper temperature; but this results in a higher temperature in the cold body and hence may not be desirable.

Obviously with given upper temperature T_1, the lower the temperature (T_2) of the cold body is maintained the smaller will be the C.o.P.; and, with given lower temperature (T_2), the lower the temperature of the hot body receiving the heat, the larger will be the value of the C.o.P.

Theoretical Values of Coefficient of Performance.

(i) For purposes of comparison with real refrigerating machinery the ideal reversible refrigerator already described is very useful although its theoretical performance can never be even closely approximated by a real machine. The case is very similar to that of engines where the perfect Carnot engine is used as a measure of perfection although practically unattainable.

(j) In order that numerical comparisons may be made later, several values of the coefficient of performance will now be obtained for a Carnot cycle refrigerator. The formula previously given can be put in more convenient form for this pur-

pose in the following way. It was shown on page 82 that for a Carnot engine the net work is

$$\Delta E = RT_1 \log_e r - RT_2 \log_e r$$
$$= (T_1 - T_2)\, R \log_e r,$$

and that $\Delta Q_2 = RT_2 \log_e r$.

Then, for this case,

$$\text{C.o.P.} = \frac{\Delta Q_2}{\Delta E} = \frac{RT_2 \log_e r}{(T_1 - T_2)\, R \log_e r} = \frac{T_2}{T_1 - T_2}. \quad . \quad . \quad (538)$$

Values of the C.o.P. can be easily obtained by substitution of assumed temperatures in the last term.

A very common case would be a machine which theoretically withdrew heat from a cold body at 18° F. and discharged it to

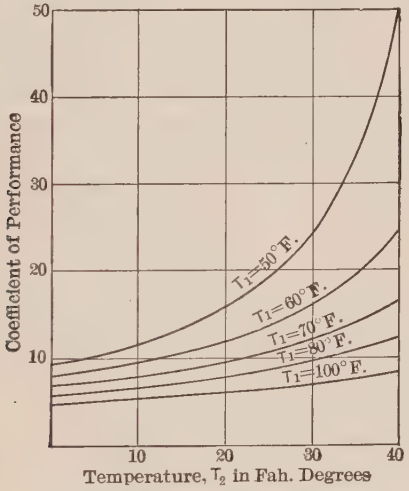


Fig. 474.

a hot body (cooling water) at a temperature of, say, 50° F. The coefficient of performance in this case would be

$$\frac{\Delta Q_2}{\Delta E} = \frac{T_2}{T_1 - T_2} = \frac{18 + 460}{(50 + 460) - (18 + 460)} = 15, \text{ approx.}$$

(**k**) The results obtained by varying the two temperatures are shown by the curves in Fig. 474, in which each curve is drawn for a certain upper temperature T_1 and shows by its rise toward the right the increase in the value of the coefficient of performance with rise of the temperature T_2.

352. The Air Refrigerating Machine. (a) Any gas, not liquefiable at ordinary temperatures, may be used as the working substance, or refrigerant, in commercial refrigerating machines, but air is the gas most commonly used. This material has the advantages of being readily procurable, non-poisonous and can be brought into actual contact with food stuffs and such, which are to be cooled or kept cool without detriment to the latter.

(b) In the ideal machine the same charge of air would be used continuously and the entire operation would be carried out in a single cylinder. In practice it is found more convenient to use separate organs to perform different functions during the cycle and it is generally found best to discharge the air used in each cycle and to draw in a fresh supply for the next. It will be noted that this parallels the conditions met with in most real engines.

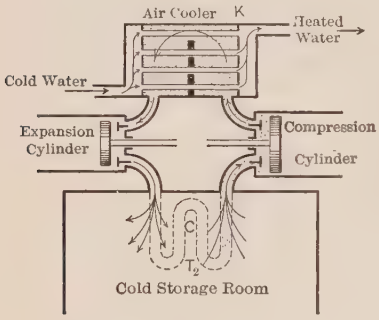


Fig. 475.

(c) An idealized refrigerating machine using a unit weight of air as a working substance is shown diagrammatically in Fig. 475, the apparatus consisting of a compression cylinder, an air cooler and an expansion cylinder.

The compression cylinder (without clearance) draws cold air from the cold-storage room at atmospheric pressure and at a constant temperature T_2, according to the constant pressure line *ab* in Fig. 476. The air is then compressed adiabatically, as shown by *bc*, and discharged at the higher pressure P_1 with a temperature T_1, higher than T_2, and specific volume *dc*. If the delivery pressure is sufficiently high, the temperature attained may be greater than that of available cooling water so that the air may be cooled by discharging it into the cooler through which this water is circulated. By assuming the volume of the cooler to be very large, the reduction of temperature may be assumed, without sensible error, to take place at constant pressure; hence, the delivery to

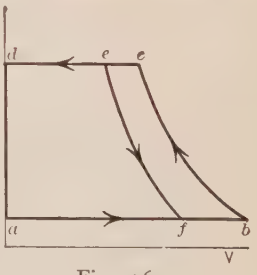


Fig. 476.

the cooler is shown by the line *cd* in the figure, but when cooled the volume of the air is *de* at this same pressure.

Continued operation of the compressor cylinder would result in continued duplication of the cycle *abcd*.

(d) The expansion cylinder running at the same speed as the compressor can be imagined as receiving from the cooler exactly the same weight of air per cycle as is delivered by the compressor. This air will be admitted according to the constant-pressure line *de*, in Fig. 476, and its adiabatic expansion will be according to line *ef*, bringing the material back to the initial pressure P_2 but with a temperature lower than the original temperature T_2, for, according to Charles' law,

$$T_f/T_b = T_f/T_2 = V_f/V_b,$$

from which

$$T_f = T_2 (V_f/V_b) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (539)$$

This cooled air at temperature T_f can then be discharged to the cold-storage room to balance heat leaking into it through the walls or brought in by fresh produce. It is only necessary to properly regulate the quantity of air handled and the temperature at which it is returned, to maintain any desired temperature (within limits) in the cold-storage room.

Power Required.

(e) The work consumed by the compressor is obviously shown by the area *abcd* and that made available in the expansion cylinder is similarly shown by the area *defa*. The net work required per cycle is then only *fbce* if the expansion and compressor pistons be connected together. This work can be expressed in terms of temperatures, pressures and volumes by the equations given in Chap. VIII.

Refrigerating Effect and Coefficient of Performance.

(f) The net refrigerating effect, that is, the heat ΔQ_2 removed from the cold room per cycle, is obviously the difference between the heat in the air as it leaves and that in the same air when it returns. That is,

$$\Delta Q_2 = WC_p (T_b - T_f), \quad . \quad . \quad . \quad . \quad . \quad (540)$$

in which

W = weight of air per cycle,
C_p = specific heat of air at constant pressure,
T_b = temperature of air leaving cold room, and
T_f = temperature of air returning to cold room.

(g) In a similar way the heat rejected to the water must be

$$\Delta Q_1 = WC_p(T_c - T_e). \quad . \quad . \quad . \quad . \quad . \quad (541)$$

Since $\Delta E = \Delta Q_1 - \Delta Q_2$, it follows that

$$\Delta E = WC_p(T_c - T_e) - WC_p(T_b - T_f), \quad . \quad (542)$$

and from these values

$$\text{C.o.P.} = \frac{\Delta Q_2}{\Delta E} = \frac{WC_p(T_b - T_f)}{WC_p(T_c - T_e) - WC_p(T_b - T_f)}$$

$$= \frac{T_b - T_f}{(T_c - T_e) - (T_b - T_f)}. \quad . \quad . \quad . \quad . \quad . \quad (543)$$

This value may be further simplified as follows: Inverting both sides of the equation gives

$$\frac{1}{\text{C.o.P.}} = \frac{(T_c - T_e)}{T_b - T_f} - 1,$$

and since from the adiabatic relation

$$\frac{T_c}{T_b} = \frac{T_e}{T_f} = \frac{T_c - T_e}{T_b - T_f},$$

$$\frac{1}{\text{C.o.P.}} = \frac{T_c - T_b}{T_b} = \frac{T_e - T_f}{T_f}$$

and

$$\text{C.o.P.} = \frac{T_b}{T_c - T_b} = \frac{T_f}{T_e - T_f}. \quad . \quad . \quad . \quad (544)$$

Comparison with the Reversed Carnot Cycle.

(h) A Carnot cycle refrigeration would work between the temperatures T_b, which is the highest temperature of the cool material, and T_e, which is the lowest temperature of the warm material. Its coefficient of performance would therefore be

$$\text{C.o.P.} = \frac{T_b}{T_e - T_b}.$$

Since $(T_e - T_b)$ is less than $(T_c - T_b)$ in Eq. (544), it follows that even in the ideal case the real refrigerating machine described must have a lower coefficient of performance than that obtained with the reversed Carnot cycle.

The difference is due to the use of two irreversible, variable-temperature, constant-pressure processes which in the real case increase the temperature range. The temperature of the air

discharged to the cooler must be so high that the water used can remove heat from it, finally bringing the air down to a value approaching that which it had when entering the compressor cylinder.

Similarly, the air is cooled during expansion to a temperature considerably lower than that of the cold room, and when introduced into that room it is heated irreversibly until it finally attains the temperature existing there.

Practical Modifications.

(i) In a real machine operating on the cycle just discussed, there will be clearance and valve losses in both cylinders, friction throughout the mechanism, and heat losses to and from the working substance as it passes through the apparatus with temperatures different from those of surrounding bodies. These will all increase the size of machine and the amount of work necessary to produce a given amount of refrigeration.

(j) In practice it is customary to water jacket the compressor. This makes the compression line less steep, i.e., intermediate between the adiabatic and the isothermal, and proportionally reduces the amount of work required. It also leaves less heat to be removed in the cooler and makes possible the use of a smaller vessel for that purpose. It is therefore decidedly advantageous.

Actual Coefficient of Performance.

(k) Both in the ideal and actual cases, the coefficient of performance of air refrigerating machines is very poor in comparison with machines using vapors such as ammonia. The use of air machines is therefore dictated by convenience rather than by economy of power.

For average cold-storage conditions, in temperate climates for instance, the coefficient of performance of a Carnot cycle refrigerator is about 9 to 10. The coefficient of the ideal air machine (Eq. (544)) is only about 1.5 to 2; and in the real machine it is generally, if not always, below 0.75, as determined by test.

353. Vapor Compression Process of Refrigeration. (a) It was shown in the previous section that the air machine there described was considerably handicapped by the cycle on which

it operated, its theoretical coefficient of performance being necessarily much lower than that of the ideal Carnot cycle refrigerator because the constant pressure reception and rejection of the heat are not reversible processes with gases.

(b) By using a liquid and its vapor as the working substance, instead of a gas, a much better performance can be obtained because the constant-pressure processes for saturated vapors and their liquids are reversible isothermal ones. It therefore follows that with such working substances the same sort of machine as that just described would in the ideal case operate on a Carnot cycle which would give the best performance possible.

(c) Fig. 477 can be used for the purpose of developing this cycle by assuming the discharge pipe of the expansion cylinder and the inlet pipe of the compressor connected by a coil, as shown dotted by C in Fig. 475, so that the entire system is "closed." This coil may be regarded as immersed in the material to be cooled.

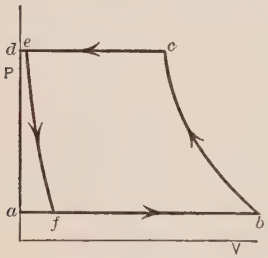


Fig. 477.

Imagine the ideal compression cylinder to draw in a charge of mixed saturated vapor and its liquid, at temperature T_2 from this coil, as shown by the line ab in Fig. 477. The return stroke of the piston will result in adiabatic compression to the point c, and, with a properly chosen quality at b, the material can be brought to the condition of dry saturation at c.

From c to d the working substance is driven into the cooler which now acts as a condenser reducing all of the vapor to the liquid form with volume de. The liquid may then be admitted to the expansion cylinder, as shown by the line de, expanded adiabatically to f and discharged along the line fa into the assumed coil C, where it may be vaporized wholly, or partly, at temperature T_2, at the expense of heat in the material surrounding the coil. After this it may be readmitted to the compressor and the cycle repeated.

(d) So far as cycle is concerned the operations outlined have resulted in the generation of the reversed Carnot cycle $fbce$. So far as heat is concerned they have resulted in the removal of heat ΔQ_2 from the cooler substance during vaporization at tem-

perature T_2 and in the surrender of a larger amount of heat ΔQ_1 to the warmer substance (condensing water) at the temperature T_1.

Tϕ-Diagram of Vapor Process.

(e) The Tϕ-changes of ammonia vapor and its liquid, in an ideal case, are shown in Fig. 478 in which points are lettered to correspond with those of Fig. 477. The liquid line and the saturation line have been added to the diagram.

From this diagram it can be seen that the heat absorbed from the cooler body is $r_2 (x_b - x_f)$ and that discharged to the warmer body is r_1. The work required in B.t.u. per pound of substance is therefore

$$\Delta E = r_1 - r_2 (x_b - x_f). \quad . \quad (545)$$

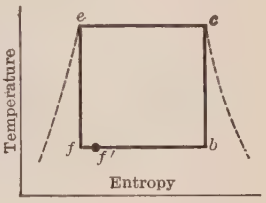


Fig. 478.

The mixture of liquid and vapor is cooled during the expansion *ef* by the giving up of heat to cause partial vaporization as indicated.

Practical Modifications of Vapor Compression Process.

(f) In any real case the expansion cylinder would be very small in comparison with the compression cylinder, and the work done by it would be practically negligible. It has come to be regarded as more of an incumbrance than a benefit and is commonly omitted in real machines. In its place is substituted an "*expansion valve*," as X in Fig. 479. This is merely a throttle valve through which the working substance can flow from the high pressure of the condenser to the low pressure of the coil.

(g) This flow is an adiabiatic process but is not reversible and hence is not isentropic. It is not represented by the line *ef* of Fig. 478, but by some line starting at *e* and terminating on the line *fb* at some point f' to the right of f. The entropy increases and the possible refrigeration effect decreases because the energy which would have been given up as external work during isentropic expansion here remains associated with the substance giving it the higher quality x_f', instead of x_f. The heat which can be absorbed from the body to be cooled is then only $r_2 (x_b - x_f')$ instead of $r_2 (x_b - x_f)$.

In real cases the difference is so small that it is negligible in comparison with the increase in mechanical efficiency and ease

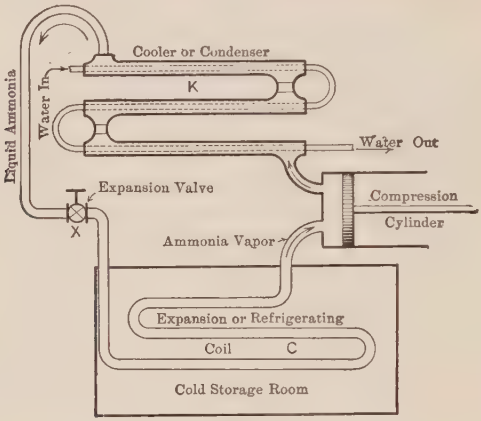


Fig. 479.

of operation, and with the decrease in first cost and operating expense.

Actual Coefficient of Performance.

(h) The great majority of vapor compression machines operate with ammonia vapor for their working substance. Such machines give a coefficient of performance of from 5 to 7 under conditions which give a coefficient of 9 to 10 for the ideal Carnot cycle. In comparison with the values given for air machines these performances are very much higher and it is doubtful if the ammonia machines can be greatly improved.

354. Relative Advantages of Different Vapors. (a) While most vapor refrigerating machines use ammonia this material is not the only one available. For plants used aboard ship carbon dioxide is often preferred and many stationary machines have been operated with this substance and with sulphur dioxide. Other materials, including water, have been used.

(b) The choice of ammonia as the common working substance is decided largely by practical considerations, though it so happens that certain thermodynamic properties would lead to the same choice. The most important considerations are probably those of volume and pressure.

(c) The actual volume of working substance required to cause a given amount of refrigeration determines the size of machine required. The larger the machine, the greater the friction losses if other things are equal. Since all friction must eventually result in the generation of heat the refrigerating effect will be diminished thereby. Bulk is therefore undesirable because of cost of machines, cost of power to operate and loss of refrigerating effect by friction.

The pressure is important in two ways. Some available substances have vapor pressures below atmospheric when at the temperatures common in refrigeration. Their use would mean the maintenance of a vacuum within the refrigerating machine which is by no means a simple matter because of difficulty with air leakage. Other substances have vapor pressures so high that they can be used only with great difficulty.

(d) Ammonia is quite satisfactory both as regards bulk and pressure. More than twice the bulk of sulphur dioxide is required for the same refrigerating effect, and between 300 and 400 times the bulk of water vapor. Carbon dioxide requires only about one-quarter the bulk of ammonia vapor but, as will be seen, is handicapped by enormously high pressures.

The pressure of water vapor is entirely below atmospheric at refrigerating temperatures, while that of sulphur dioxide is below for the lowest temperatures and only slightly above for the highest temperatures.

The pressure of ammonia vapor varies from about 20 or 25 pounds to something below 200 pounds, while that of carbon dioxide varies from about 300 to 1000 pounds per square inch.

It is obvious that the best commercial balance is struck when ammonia is adopted, excepting in cases where an ammonia leak might cause very serious difficulties.

(e) In the case of real machines there is also another point which must be considered and which is more of a thermodynamic nature. Where an expansion valve is substituted for the expansion cylinder, the working substance brings into the refrigerating coil heat which in the ideal case would have been converted into work and used in driving the machine. Obviously any heat brought into the coil by the working substance itself means just so much less heat to be abstracted from the surroundings to cause evaporation, hence there will be an equal reduction in the refrigeration.

That material which brings in relatively the smallest amount of heat in this way will be the most desirable if other things are equal.

The amount of heat under consideration is that in the liquid at the end of the liquefaction process, that is, it is the quantity when the working substance is at the higher temperature T_1 which is above that in the same liquid at the lower temperature T_2. It is therefore equal to $C\ (T_1 - T_2)$ in which C is the specific heat of the liquid. The larger this value in proportion to the latent heat of vaporization at the temperature T_2, the poorer the material for use in a vapor compression machine having an expansion valve.

From this point of view, water is the best of the materials cited as possibilities and ammonia comes next, carbon dioxide being the worst of all; thus, ammonia forms a good commercial compromise.

355. The Ammonia Absorption Process. (a) The vapor compression machine operates (i.e., refrigerates) because the process of vaporization requires a supply of heat from external sources and the process of liquefaction yields heat to external media. Any device or machine which can bring about such alternate liquefaction and vaporization can be used as a refrigerating machine.

(b) The so-called absorption refrigerating machine carries through these two processes in a manner analogous to that of the compression machine but by entirely different means. It is illustrated diagrammatically in Fig. 480 and operates in the following way:

(c) The generator contains a strong solution of ammonia in water, and the ammonia is driven off from this solution at high temperature and pressure, by the heat supplied by the steam coils shown at S. The vapor, under this pressure, enters the condenser K in which it is liquefied, as in the previous case. It then passes through the expansion valve X and evaporates in the refrigerating coils C as before.

Leaving the refrigerating coils as vapor, it enters the absorber A at low pressure and low temperature and is absorbed by water to form a strong solution which, by a pump P, is delivered to the generator to displace that which has given up ammonia

vapor under the action of heat, and which is then returned to the absorber.

(d) The absorber, pump and generator together correspond to the compressor of the previous type. The action in the ab-

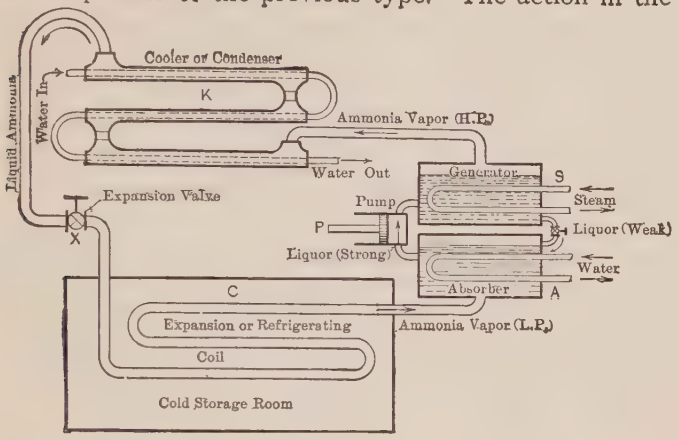


Fig. 480.

sorber corresponds to the charging operation of the compressor; the action of pump and generator corresponds to the compression and discharge.

Coefficient of Performance of Absorption Machines.

(e) No mechanical power, except the small amount for pump P, is supplied such a machine, the absorption of heat at a low temperature following from the supply of heat at a high temperature. The coefficient of performance cannot therefore be obtained as in previous cases. If, however, it is considered as the quotient found by dividing the heat absorbed by heat supplied to cause that absorption, a ratio is obtained which may be used in the same way as the coefficient of performance. Comparing such ratios with those for ideal refrigeration operating on a reversed Carnot cycle it is found that the absorption machine has a coefficient of performance of about one-eighth to one-tenth that of the ideal.

(f) It was shown in Sect. 353 (h) that for the compression process the coefficient is about seven-tenths of the ideal and it would seem from this that the absorption machine should give a very poor commercial result. It should, however, be observed

that the coefficient for the compression machine was based upon the energy supplied the compressor and not upon the heat supplied the plant which generated that energy.

To make the two results comparable the value of 0.7 for the compression machine must be multiplied by the thermal efficiency of the plant on the basis of developed horse power. When this is done the two types are more nearly on an equal footing, with the absorption machine giving the better performance for wide temperature ranges, excepting when a very efficient plant is used to drive the compression machine.

356. Rating of Refrigerating Machines. (a) Refrigerating machines are generally used in practice for the purpose of maintaining a cold atmosphere in "cold-storage rooms" or for the making of ice. The ammonia machines generally achieve both results indirectly by cooling brine and pumping the brine to the point at which heat is to be absorbed.

(b) No matter what use is made of the refrigerating machine or how it operates, it is rated on *ice-melting capacity* in pounds, or tons, per unit of time. To melt one pound of ice at 32° F. to water at the same temperature requires approximately 144 B.t.u.

A machine which could absorb from the cold body a quantity of heat equal to 144 B.t.u. per hour would have an ice-melting capacity of one pound per hour. The capacity is generally expressed in tons per twenty-four hours, thus this machine would have an ice-melting capacity of $(1 \times 24) \div 2000 = 0.012$ ton, approximately.

Ice-melting capacity has no direct connection with *ice-making* capacity. When making ice the water from which it is made must first be cooled to freezing temperature, the ice then formed, and generally reduced to a temperature considerably below 32° F. As a result, the ice-making capacity of a machine is generally only about one-half of its ice-melting capacity.

PROBLEMS.

CHAPTER II.

1. Assuming the specific heat of water constant and equal to unity, how many B.t.u. are required to raise the temperature of 1 lb. of water from 32° F. to 212° F.?

2. Under the same assumptions as above, how many B.t.u. must be abstracted to lower the temperature of 20 lbs. of water from 212° F. to 32° F.?

3. If 33,000 ft.-lbs. of mechanical energy are completely converted into heat energy, how many B.t.u. result?

4. If mechanical energy is made available at the rate of 33,000 ft.-lbs. per minute for 1 hour it is said that 1 horse-power hour has been made available. What is the energy equivalent of 1 horse-power hour in thermal units?

5. Find the weight of water (specific heat = 1) which will have its temperature doubled by the addition of 180 B.t.u., the final temperature being 120° F.

6. Find the change of temperature of 12 oz. of lead (specific heat = 0.0314) when 4 B.t.u. are added.

7. Assuming no loss by radiation, how much electrical energy in terms of thermal units would be required to raise the temperature of a copper wire one mile long and weighing 0.3 lb. per foot through a range of 10 degrees? The specific heat of copper is 0.095.

8. If 130 B.t.u. raise the temperature of 10 lbs. of cast iron 100 degrees, what must be the specific heat of this material?

9. Assume the specific heat of wrought iron as 0.113, the specific heat of water as 1.0 and the weight of water as 62.5 lbs. per cu. ft. Find the increase in temperature of 2 cu. ft. of water when a common temperature of 45° results from putting into the water a piece of iron weighing 10 lbs. and at a temperature of 1000° F.

10. A winch is used in lowering a load of two tons a vertical distance of 50 ft. The load is lowered by means of a friction brake which prevents the attainment of too high a speed and which brings the load to rest just as it reaches the end of the 50-ft. drop. It takes one minute to lower the load. Neglecting friction of bearings and similar losses, how much heat must be radiated by the mechanism of the brake and winch? How many horse power must be absorbed by the brake?

11. An electric motor receives electrical energy, converts part of it into heat within itself and delivers the remainder at the pulley as available mechanical energy. A certain motor delivers, in this way, 20 horse power (1 h.p. = 33,000 ft.-lbs. per min.) and converts into heat 15 per cent of all the energy supplied it. How much heat must this motor dissipate per hour? How many ft.-lbs. of energy must be supplied it per minute?

12. Assume yourself called upon to investigate the claims made for a piece of mechanism with the following characteristics. It receives no energy of any kind excepting that given it by a driving belt which supplies 250,000 ft.-lbs. per minute. It is claimed that the mechanism gives out or makes available 400 B.t.u. per minute. Would you make the investigation? Why?

13. What is the largest amount of heat energy which the mechanism operating as in problem 12 could make available per minute in an ideal case? Could it do this in practice? Why?

14. Assume yourself called upon to investigate the claims made for a piece of mechanism with the following characteristics. It is supposed to receive no energy of any kind excepting 300 B.t.u. per minute and is supposed to make available 240,000 ft.-lbs. of mechanical energy in the same time. Would you make the investigation? Why?

15. Assume that the mechanism in problem 14 above is supposed to receive only 300 B.t.u. as before and that in the ideal case (neglecting friction, radiation, conduction and similar losses) it is supposed to deliver 233,400 ft.-lbs. in the same time. Would you make the investigation? Why?

16. Assume that the value of the variable specific heat C of a subtance is given for temperature t by the equation

$$C = 0.5 + 0.02\,t.$$

Find the total heat required to raise the temperature of 12 lbs. of the material from 50° to 100° F.

17. An engine receiving 300 B.t.u. per minute and no other energy of any kind rejects to a cold body an amount of energy equal to 150 B.t.u. per minute. If there are no friction or similar losses, what would be the amount of mechanical energy in ft.-lbs. made available per minute?

18. If the engine operating as in the first part of prob. 17 above loses in friction and radiation 10 per cent of the energy which would otherwise be made available, what will be the amount of mechanical energy in ft.-lbs. made available per minute?

19. A factory building is being designed. Calculations from the radiating surface of the building, character of that surface, location, direction of winds, etc., indicate that about 400,000 B.t.u. per hour must be liberated within the building to keep the temperature up to 65° F. The heating engineer desires to keep the cost of the heating equipment down to a low figure and believes that he can do so by allowing for heat generated by friction of the moving mechanisms within the factory. He discovers that 100 horse power (1 h.p. = 33,000 ft.-lbs./min.) are to be continuously supplied the factory by means of an electric motor and that all of this power will be consumed within the factory. The motor has an efficiency of 85 per cent. What allowance can the heating engineer make on theoretical grounds?

20. In the manufacture of a certain chemical compound it is necessary to stir and mix a rather heavy liquid in a large vat. If the temperature of the liquid rises above a certain value it is apt to cause a violent explosion. The formation of the compound causes the absorption of 20,000 B.t.u. per hour and the vat is so arranged that 25,000 B.t.u. can be carried away per hour under all conditions by means of a water jacket and loss to the surrounding atmosphere. How many foot-pounds of energy could be supplied the stirring apparatus per hour without causing a dangerous rise of temperature?

CHAPTER IV.

1. An ideal gas occupies a volume of 17 cu. ft. at a pressure of 1500 lbs. per sq. ft. and a temperature T. What will be its volume at a pressure of 2000 lbs. per sq. ft. and at the same temperature?

2. A gas has its volume halved by an increase of pressure at constant temperature. The initial pressure was 3000 lbs. per sq. ft.; what is the final pressure?

3. A gas with an initial pressure of 4500 lbs. per sq. ft. is contained in a water-jacketed cylinder, the jacket being so connected with a water system that the temperature within it is always 60° F. The cylinder is fitted with a frictionless piston which can be moved in or out as desired. The piston is moved very slowly so that the gas is maintained at the same temperature as the water jacket. At the end of a certain time the pressure of the gas within the cylinder is found to be 14.7 lbs. per sq. in. Was the piston moved in or out? What is the ratio of the final volume to the initial?

4. A balloon is filled with hydrogen gas at atmospheric pressure (14.7 lbs. per sq. in.) and at atmospheric temperature. The balloon then ascends to a point where the atmospheric pressure is only 12.7 lbs. per sq. in. but the temperature is the same as at the lower level. If the balloon is made of non-extensible material, what will be the pressure of the hydrogen gas within it? If the balloon is made of perfectly stretchable material (stretching with application of only infinitesimal forces), what will be the pressure within it? In the latter case what expansion of volume must have occurred?

5. The inner tube of a certain tire has a capacity of 854 cubic inches. How many pounds of air will it contain when filled with air at a pressure of 70 lbs. per sq. in. and a temperature of 32° F.? (One cubic foot of air at 14.7 lbs. pressure and 32° F. weighs 0.0807 lbs.)

6. What will be the increase of pressure of air in problem 5 if the temperature rises to 70° F. and the tire does not stretch during the process?

7. A closed metal tank is designed to be safe when subjected to an internal pressure of 100 lbs. per sq. in. It is used to hold compressed air and is filled with this material at a temperature of 60° F. and a pressure of 80 lbs. per sq. in. The tank stands in the sun and its contents may attain a temperature of 125° F. Assuming that the tank does not expand with temperature and pressure changes, will the designed pressure be exceeded? What temperature would have to be attained to raise the pressure of the air to the 100 lbs. for which the tank was designed?

8. A submarine boat is closed at the surface, with air content at a temperature of 80° F. and a pressure of 14.5 lbs. per sq. in. After sinking beneath the surface the temperature of the air drops to 40° F. If the hull has not changed size during the temperature change what must be the pressure of the air under the submerged conditions?

9. Assuming that the men and machinery in the boat of problem 8 radiate enough heat to maintain a temperature of 60° within the boat what will the air pressure be?

10. A quantity of gas occupies a volume of 10 cu. ft. at a pressure of 5000 lbs. per sq. ft. and a temperature of 70° F. What will its volume be at a pressure of 7000 lbs. per sq. ft. and a temperature of 100° F.?

11. A quantity of gas occupies a volume of 15 cu. ft. at a pressure of 40 lbs. per sq. ft. and a temperature of 60° F. What will be its volume at a temperature of 70° F. and a pressure of 30 lbs. per sq. ft.?

12. A quantity of gas occupies a volume of 10 cu. in. at a pressure of 7000 lbs. per sq. ft. and a temperature of 50° F. What will be its volume at atmospheric pressure (14.7 lbs. per sq. in.) and a temperature of 70° F.?

13. The value of R for a certain gas is 55. One pound of this gas occupies a volume of 12.8 cu. ft. at a pressure of 14.7 lbs. per sq. in. What is the temperature of the gas? What will be the volume of two pounds of this gas if pressure and temperature (ordinary Fahrenheit scale) are doubled?

14. Three pounds of air are enclosed in a nonexpansible vessel. The pressure is 20 lbs. per sq. in. and the temperature is 80° F. The value of R is 53.34. What is the volume content of the vessel? What will be the pressure of the air if its temperature is increased to 180° F.?

15. One-half pound of nitrogen gas is contained in a cylinder fitted with a piston. The temperature of the gas is 80° F. and its pressure is 40 lbs. per sq. in. The piston moves out until the volume of the gas has doubled and it is then found that its pressure is 20 lbs. per sq. in. What must the temperature have become? (R for nitrogen = 55.16.)

16. A certain gas is collected over mercury and measured. It is found to have a volume of 10 cu. in. at a pressure of 14.6 lbs. per sq. in. and a temperature of 60° F. The gas is then passed through a reagent which absorbs part of it and the remainder is collected over mercury and measured. It measures 6 cu. in. at the same pressure as before but the temperature has changed to 70° F. between the two measurements. What percentage of the original volume was absorbed by the reagent?

17. The products of combustion from a boiler reach the base of the stack

at a temperature of 500° F. At the top of the stack their temperature is only 200° F. Neglecting the slight pressure change which would occur during the ascension, determine the relative values of the cross-sectional areas at top and bottom of the stack to give equal gas velocities at the two points.

18. An air compressor draws into its cylinder a charge of air at a pressure of 13.5 lbs. per sq. in. and a temperature of 60° F. It compresses this air to a volume equal to one-quarter of its original value and the pressure attained is 60 lbs. What must be the final temperature of the air?

19. A diving bell is to be used for executing certain work under water. It is made in the form of a flat-ended cylinder open at the bottom. The inside diameter is 12 ft. and the inside height is 14 ft. The men and tools accommodated within the bell occupy a cubical content of 120 cu. ft. If the bell is lowered into the water when atmospheric pressure is 14.7 lbs. per sq. in. and temperature is 60° F., how far below the surface can the bottom of the bell be lowered if the water has a temperature of 40° F., weighs 62.5 lbs. per cu. ft., and is not to rise to a height of more than 4 ft. from the bottom of the bell? Assume that men and tools remain entirely within the air space, that they do not change volume with pressure change; that the air within the bell acquires the same temperature as the surrounding water.

20. A quantity of heat equal to 1000 B.t.u. ($= \Delta Q$) is given to an ideal gas maintained at constant volume. What are the numerical values of the several terms in the equation $\Delta Q = \Delta S + \Delta I + \Delta E$?

21. Two pounds of a gas with $C_v = 0.1662$ and $C_p = 0.2317$ are heated at constant volume from a temperature of 60° F. to a temperature of 80° F., and then at constant pressure to a final temperature of 100° F.

(*a*) How much heat is supplied to the gas?

(*b*) What is the value of ΔS for each part of the process?

(*c*) What is the value of ΔE for each part of the process?

22. The true specific heat of a certain gas is 0.1733 in thermal units. The value of R is 55.16, what is the value of C_p for this gas?

23. Three pounds of an ideal gas are heated until 200 B.t.u. per lb. have been supplied it. During the process the gas expands and 133,038 ft.-lbs of work are done by it. Find the value of ΔS.

24. A balloon is filled with hydrogen gas at a pressure of 14.7 lbs. per sq. in and a temperature of 60° F. The balloon is spherical in shape and has an internal diameter of 25 ft. At a later time it is found that the pressure of the gas within the balloon is only 0.95 of the original value but that the temperature is the same as before. What fraction of the original weight of gas must have escaped if the dimensions of the balloon have not changed? How much heat would have had to be removed to cause the pressure to fall to the same extent if no leakage occurred? (1 lb. of hydrogen at 32° F. and 14.7 lbs./sq. in. occupied a volume of 178 cu. ft. $R = 766.5$; $C_p = 3.41$.) What would be the final temperature?

25. Ten pounds of air (specific volume at 32° and 14.7 lbs. = 12.387 cu. ft.) are contained in a receiver at a temperature of 55° F. and a pressure of 100 lbs. per sq. in. Air leaks out until at a later time the pressure in the receiver is found to be only 40 lbs. per sq. in. with a temperature of 50° F. What weight has leaked out?

26. An air receiver has a factor of safety of 5 when filled with air at a pressure of 200 lbs. per sq. in. and a temperature of 100° F. What amount of heat would have to be supplied the air to reduce the factor of safety to 2.5 on the assumption that the cubical content of the receiver remains constant with changing temperatures. ($C_p = 0.2374$; $\gamma = 1.4037$.) Assume vol. = 20 cu. ft.

27. The value of R for a certain gas is 34.9. How much external work will be done by five pounds of this gas if its temperature is raised from 50° F. to 150° F. at constant pressure?

28. The value of C_p for a certain gas is 0.23 and the value of γ is 1.39. What volume must this gas occupy when at a temperature of 60° F. and subjected to a pressure of 150 lbs. per sq. in.? Assume 5 lbs. of gas.

29. A certain gas with molecular weight equal to 28 occupies a volume of

12.8 cu. ft. per lb. What volume will another gas with molecular weight of 32 theoretically occupy when at the same temperature and pressure?

30. A certain gas with molecular weight of 44 weighs 0.1224 lbs. per cu. ft. at standard conditions. Another gas has a molecular weight of 26. What is its theoretical density under the same conditions of temperature and pressure?

31. A water pump running at 50 strokes per minute delivers 1 cu. ft. of water per stroke. An air chamber is to be fitted to this pump of such size that the discharge pressure on the pump shall vary from 100 lbs. per sq. in. at the beginning of the stroke to 150 lbs. per sq. in. at the end of the stroke if all the water delivered during one stroke must be accommodated in the air chamber. The temperature of the water and of the air in the chamber remain constant at 60° F. and no air is absorbed by the water. (*a*) What must be the volume of the air chamber if R for air is equal to 53.3? (*b*) Would there be any economic advantage in using a gas with $R = 96$?

32. Assume that gas is to be used for the doing of external work by being heated at constant pressure through a certain temperature range. If a large number of gases are available but only one pound of any one gas can be used, would you select the gas having the lowest or highest value of R if maximum amount of work was a consideration? Why? What other property of the gases would you consider if size of machine was also of importance? Why? Assuming that it is desired to determine the gas which would give the maximum amount of work with the smallest machine, how would you proceed?

CHAPTER V.

1. (*a*) How much work can be done by two pounds of air expanding at a constant pressure of 50 lbs. per sq. in. to twice the original volume if the initial temperature is 50° F.? (*b*) What will be the final temperature? (*c*) How much heat will have to be supplied the gas? ($C_p = 0.2374$.)

2. One-half pound of nitrogen is inclosed in a cylinder fitted with a frictionless piston. When the gas has a temperature of 100° F. the pressure upon the piston is 25 lbs. per sq. in. It is desired to abstract 20 B.t.u. from the gas without changing its pressure. What will be the temperature drop and how much must the volume be decreased? ($C_p = 0.2438$.)

3. A vessel with a capacity of 5 cu. ft. is filled with air at a pressure of 125 lbs. per sq. in. when at a temperature of 60° F. It is desirable to lower the pressure to 50 lbs per sq. in. What amount of heat will have to be abstracted and what will be the final temperature of the gas, assuming that the vessel does not change in size with change of temperature? ($R = 53.34$; $\gamma = 1.4037$.)

4. A cylinder permanently closed at one end is fitted with a frictionless piston and stands vertical upon its closed end in a vacuum. It holds a volume of 1 cu. ft. of gas at a temperature of 75° F., and a pressure of 20 lbs. per sq. in., the pressure being maintained by the weight of the piston and superposed discs of metal. The temperature is raised to 150° F. and the piston is prevented from rising by additional weights placed upon it. (*a*) What percentage of the original weight of piston and discs must be added? (*b*) If no additional weights had been added how much external work would have been done by the expanding gas?

5. How much work must be done to compress 100 lbs. of air isothermally from a pressure of 13 lbs. per sq. in. at a temperature of 70° F. to a pressure of 110 lbs. per sq. in.? (Spec. vol. at 32° F. and 14.7 lbs. = 12.387.)

6. How much heat must be absorbed during the process assumed in problem 5 above?

7. One pound of air expands isothermally in a cylinder behind a frictionless piston. The initial pressure is 100 lbs. per sq. in., the initial temperature is 50° F., the ratio of expansion is 5. (Spec. vol. at 32° F. and 14.7 lbs. = 12.387.) (*a*) What amount of work will the gas do upon the piston? (*b*) How much heat will have to be supplied the gas during the expansion?

8. Find the work done by 0.5 lb. of carbon dioxide expanding isothermally at 100° F. from an initial pressure of 100 lbs. per sq. in. to a final volume of 10 cu. ft. (Spec. vol. at 32° and 14.7 lbs. = 8.1 cu. ft.)

9. Air is compressed at constant temperature from a volume of 60 cu. ft. and a pressure of 14.7 lbs. per sq. in. to a volume of 12 cu. ft. Find (*a*) final pressure, (*b*) heat removed, and (*c*) work done upon the gas.

10. Find the work done by 4 lbs. of air expanding isothermally from a pressure of 100 lbs. per sq. in. to 20 lbs. per sq. in., the final volume being 80 cu. ft.

11. Five pounds of air expand isothermally from a pressure of 120 lbs. per sq. in. to a final pressure of 20 lbs. per sq. in., the work done being 248,500 ft.-lbs. ($R = 53.3$.) Find (*a*) initial volume, (*b*) final volume, (*c*) initial temperature, and (*d*) heat supplied.

12. The volume of 1 lb. of air at 32° F. and 14.7 lbs. per sq. in. = 12.387 cu. ft. (*a*) Find work done during the isothermal expansion of one pound of air from 100 lbs. per sq. in. to 20 lbs. per sq. in. at 100° F. ($\log_e 5 = 1.61$.) (*b*) Find initial and final volumes. (*c*) Find value of R.

13. A given weight of gas occupies 3.09 cu. ft. and is under a pressure of 200 lbs. per sq. in. It expands isothermally, the ratio of expansion being 3. Find (*a*) final volume, (*b*) ft.-lbs. of work done, and (*c*) B.t.u. necessary to do this work.

14. Atmospheric pressure at sea level on a certain day is 14.7 lbs. per sq. in. and on a certain mountain it is 12.5 lbs. per sq. in. An air compressor at each place compresses isothermally 100 cu. ft. of air per minute (measured at existing atmospheric pressure and same temperature in each case) to a pressure of 80 lbs. per sq. in. How much work is done upon the gas in each case? What is the difference, and what per cent of the smaller quantity is used in excess in the less favorable location?

15. If, in the preceding problem, each compressor had raised the pressure to five times atmospheric pressure at its own location, how would the quantities of work compare?

16. A gas with $\gamma = 1.4$ expands adiabatically from an initial volume of 10 cu. ft. and an initial pressure of 100 lbs. per sq. in. to a terminal pressure of 15 lbs. per sq. in. What is the final volume?

17. A gas with $\gamma = 1.35$ expands adiabatically from an initial volume of 0.4 cu. ft. and an initial pressure of 80 lbs. per sq. in. to a final volume of 4 cu. ft. What is the final pressure?

18. A gas with $\gamma = 1.41$ is compressed adiabatically from an initial volume of 5 cu. ft. and an initial pressure of 15 lbs. per sq. in. to a final volume of 1 cu. ft. What is the final pressure?

19. A gas with $\gamma = 1.33$ is compressed adiabatically from an initial volume of 2 cu. ft. and an initial pressure of 15 lbs. per sq. in. to a final pressure of 85 lbs. per sq. in. What is the final volume?

20. One pound of air expands adiabatically in a cylinder fitted with a frictionless piston. The initial pressure is 100 lbs. per sq. in.; the initial temperature is 50° F.; the ratio of expansion is 5. (Spec. vol. at 32° and 14.7 lbs. = 12.387; $\gamma = 1.4037$.) (*a*) What amount of work will the gas do upon the piston? (*b*) How much heat will have to be supplied the gas during the expansion? (*c*) Compare with results of problem 7.

21. Five pounds of gas with $\gamma = 1.4$ expand adiabatically from a volume of 0.2 cu. ft. and a pressure of 90 lbs. per sq. in. to a final pressure of 18 lbs. per sq. in. (*a*) What is the final volume? (*b*) How much external work is done by the gas?

22. How much work must be done upon two pounds of gas to compress them adiabatically from a volume of 2 cu. ft. and a pressure of 14 lbs. per sq. in. to a final pressure of 80 lbs. per sq. in.? The value of γ is 1.4.

23. What will be the difference in the amounts of work required to compress 5 cu. ft. of free air (air at 60° F. and 14.7 lbs. per sq. in.) to a pressure of 90 lbs. per sq. in. when the compression is adiabatic and when it is isothermal? ($\gamma = 1.4037$.)

24. Assume that an air compressor can be so arranged as to compress air either isothermally or adiabatically. It receives air at a pressure of 14 lbs. per sq. in. and compresses to a pressure of 75 lbs. per sq. in. ($\gamma = 1.41$.) (*a*) How much work would be done in compressing an initial volume of 1 cu. ft. of air by each method? (*b*) What would be the percentage of saving when using the more economical method?

25. Five pounds of gas have an initial pressure of 300 lbs. per sq. in. and occupy an initial volume of 20 cu. ft. ($C_p = 0.238$; $R = 53.3$.) (*a*) Find C_v and T_1. (*b*) If this gas is expanded adiabatically to a pressure of 150 lbs. per sq. in., what will be the numerical value of V_2, T_2 and work done?

26. One-quarter of a pound of gas with $C_p = 0.238$ and $C_v = 0.169$ is expanded adiabatically from $V_1 = 0.2$ cu. ft. and $p_1 = 300$ lbs. per sq. in. to $p_2 = 150$ lbs. per sq. in. (*a*) What are the numerical values of R, T, V_2, T_2? (*b*) What is the numerical value of work done?

27. One-quarter of a pound of air is compressed adiabatically from 13 lbs. per sq. in. and 60° F. to a pressure of 100 lbs. per sq. in. After compression its temperature is decreased to 60° F. while the volume is maintained constant. (*a*) How much heat will have to be abstracted to bring this about? (*b*) What will be the final pressure? (*c*) If the gas is now allowed to expand adiabatically to a pressure of 13 lbs., how much work can it do and how does this compare with that required in the original compression? (*d*) What will be the volume and temperature at end of expansion as in (*c*) above? (Spec. vol. at 32° and 14.7 lbs. = 12.387; $\gamma = 1.4037$; $C_p = 0.2374$.)

28. Using logarithmic cross-section paper determine the pressure exerted by a gas for each cubic foot of volume increase when expanding according to the law $PV^{1.35}$ = constant, from an initial pressure of 100 lbs. per sq. in. and an initial volume of one and one-half cu. ft. to a terminal pressure of 15 lbs. per sq. in.

29. Air is drawn into an air compressor at a temperature of 61° F. and at atmospheric pressure (14.7 lbs. per sq. in.). The flash point of the oil used to lubricate the compressor piston is 350° F. If compression is adiabatic, what pressure could be attained in the compressor if the maximum allowable temperature is 50 degrees below the flash point of the oil? (Assume $\gamma = 1.41$.)

30. An air compressor compresses adiabatically 100 cu. ft. of air per minute measured at initial conditions of 15 lbs. per sq. in. and 60° F. The final pressure is 90 lbs. per sq. in. (*a*) Find work done on air per minute. (*b*) Find final temperature. (*c*) Find weight of air compressed per minute. (Spec. vol. = 12.387 at 32° F. and 14.7 lbs.; $R = 53.3$; and $\gamma = 1.41$.)

31. The initial conditions of two pounds of gas are $p_1 = 100$ lbs. per sq. in. and $t_1 = 60°$ F. (R for this gas is 53.3 and $\gamma = 1.41$.) (*a*) How much work will be done by the gas if it expands isothermally to a final pressure of 15 lbs. per sq. in.? (*b*) If the expansion is adiabatic? (*c*) What is the percentage gain by the former method? (*d*) How is this gain purchased? (*e*) How do the final temperatures compare?

32. Power is obtained by expanding air adiabatically in an engine cylinder. It is found that when the temperature of the air drops below 32° F. the moisture which is carried by the air freezes and impairs the action of the engine. Air is received by the engine at a temperature of 60° F. and a pressure of 100 lbs. per sq. in. (Assume $\gamma = 1.41$ and assume further that the quantity of moisture present in the air is so small as to have no thermodynamic effect, i.e., all formulas may be used as though dry air only were present.) (*a*) What is the lowest pressure to which the air can expand if its temperature is not to drop below 32° F.? (*b*) To what initial temperature would it be necessary to heat the air in order that it may be possible to expand to 15 lbs. per sq. in. without dropping below the minimum allowable temperature?

33. Air is available for use in a compressed air engine at a temperature of 65° F. and at a pressure of 125 lbs. per sq. in. It is desired to preheat it at constant pressure to such a temperature that it will not drop below a temperature of 35° F. when expanded to a final pressure of 16 lbs. per sq. in. according to the equation $PV^{1.35}$ = const. How much heat will be required per pound if $C_p = 0.237$?

34. How much work will be required to compress two pounds of gas from $V_1 = 25$ cu. ft. and $p_1 = 13.5$ lbs. per sq. in. to $p_2 = 75$ lbs. per sq. in. according to the equation $PV^{1.35} = \text{const.}$? What will be the final temperature if the initial temperature is 55° F.?

CHAPTER VI.

1. Is the following process thermodynamically reversible? Why? Gas in contact with a hot body with the same temperature as the gas, receives heat from that hot body while expanding isothermally and doing work.

2. Is the following process thermodynamically reversible? Why? Gas expands isothermally and does a certain amount of work at the expense of heat received from a hot body at temperature 10 degrees higher than that of the gas.

3. Is the following process thermodynamically reversible? Why? A gas is made to expand at constant pressure by being brought into contact with a hot body.

4. Is the following process thermodynamically reversible? Why? A gas maintained at constant volume has its pressure decreased by being brought into contact with a cold body.

5. Is the following process thermodynamically reversible? Why? Gas is compressed adiabatically in a nonconducting cylinder.

6. Is the following process thermodynamically reversible? Why? Gas is compressed in a cylinder with metallic walls.

7. Is the following process thermodynamically reversible? Why? A blacksmith strikes his anvil forcibly with his hammer.

8. Is the following process thermodynamically reversible? Why? The hot gases resulting from the combustion of fuel within a boiler furnace flow up a smoke stack because their density is less than that of the atmosphere surrounding the stack. Is the process going on in the stack reversible?

9. Is the following process thermodynamically reversible? Why? A carpenter bores a hole in a piece of wood by means of a brace and bit.

10. Is the following process thermodynamically reversible? Why? A machinist cuts a thread upon a bar of metal which is rotated in a lathe. The bar and cutting tool are kept cool by a stream of soap solution. Outline the energy changes occurring and tell whether the process is reversible so far as these energy changes are concerned.

11. Assume two vessels of equal cubical content arranged as in Sect. 35 (**b**), one containing one pound of air at a temperature of 60° F. and a pressure of 100 lbs. per sq. in., the other absolutely void. (Spec. volume of air at 32° F. and 14.7 lbs. = 12.387). Assume that it is possible to open the cock between the two suddenly, to allow gas to flow from the high-pressure to the low-pressure vessel until both have the same pressure of gas, and then to close the cock suddenly so as to isolate the two bodies of gas. Assume further that the material of which the two vessels and fittings are made is absolutely impervious to heat. (*a*) At the end of the process what will be the temperature of the gas contained in the vessel originally charged with high-pressure gas? (*b*) At the end of the process what will be the temperature of the gas contained in the vessel originally void?

CHAPTER VII.

1. Find the change of entropy of 4 lbs. of a gas heated at constant pressure from a temperature of 60° F. to a temperature of 1000° F. ($C_v = 0.192$; $R = 45.3$.)

2. If 6 lbs. of air are cooled at constant volume until the final pressure is one-fourth of the initial, find the change of entropy. ($C_v = 0.169$.)

3. If 10 lbs. of air are heated at constant pressure until $t_2 = 2\,t_1$, thereby adding 1185 B.t.u. to the gas, find the change of entropy. ($C_p = 0.237$.)

4. Find the entropy change of gas which is compressed isothermally from a pressure of 14.7 lbs. per sq. in. and a volume of 60 cu. ft. to a volume of 12 cu. ft. The gas is maintained at a temperature of 80° F.

5. If 25 lbs. of carbon dioxide, having $R = 35.1$, are compressed from a volume of 50 cu. ft. to a volume of 10 cu. ft., the pressure remaining constant, find the change in entropy. ($C_p = 0.2008$.)

6. If 5 lbs. of air expand isothermally from a pressure of 100 lbs. per sq. in. to a pressure of 20 lbs. per sq. in. and a temperature of 60° F., find the change in entropy. ($R = 53.3$.)

7. If $\frac{1}{2}$ lb. of air is allowed to expand isothermally at a temperature of 70° F. until its final volume is 4 times its initial one, find its change in entropy. ($R = 53.3$.)

8. Find the entropy change of 5 lbs. of a gas which expands isothermally at 60° F. until the ratio of its final volume to initial volume is 14.8. ($C_p = 0.2008$; $C_v = 0.1548$.)

9. Find how much heat would be required to heat 4 lbs. of air at constant volume so that it would experience an entropy change of 0.468, its initial temperature being 60° F. ($C_v = 0.169$.)

10. 3 lbs. of air are compressed isothermally from a volume of 36 cu. ft. and pressure of 15 lbs. per sq. in. to a volume of 9 cu. ft. Find the change in entropy. ($R = 53.3$.)

11. At the end of the compression stroke in a gas engine cylinder, the temperature is found to be 970° abs. and the entropy change from 32° F., and same volume as at end of compression, is 0.55. After combustion at constant volume (pressure rise at const. vol.) the entropy has increased to 0.96. ($C_v = 0.16$.)

(1) What is the final temperature at the end of combustion?

(2) How much heat has been added?

12. Imagine 0.4 lb. of an ideal gas to expand in a cylinder which prevents any heat flow to or from the gas, initial pressure being 100 lbs. per sq. in. and initial volume $\frac{1}{2}$ cu. ft., find the work done when its volume has become 3 cu. ft. Find change in temperature. Find the change in entropy. ($C_p = 0.124$. $C_v = 0.093$.)

CHAPTER VIII.

1. A Carnot cycle is performed with gas as a working substance. The temperature of the hot body is 1000° F. and that of the cold body is 60° F. How much work is done per cycle if the heat supplied per cycle is 10 B.t.u.?

2. In the case of the Carnot cycle as above with higher temperature 1000° F. and lower temperature 60° F., how much work would be done per cycle if the heat rejected per cycle equals 10 B.t.u.?

3. A Carnot cycle with gas as working substance is used for the development of power. It is desired to obtain 100 ft.-lbs. of work per cycle. The heat supplied per cycle equals 0.3 B.t.u. and the temperature of the hot body is 500° F. What must be the temperature of the cold body?

4. A Carnot engine is to be used as a heat pump to remove 10 B.t.u. per cycle from a body at a temperature of 32° F. and discharge to a body at a temperature of 100° F. (*a*) How much energy will be required per cycle to operate this heat pump? (*b*) How much will be required if the upper temperature is 200° F.? (*c*) How much heat will be discharged to the hot body in each case?

5. An engine using air as a working substance, receiving heat from a hot body at temperature 1000° F. and rejecting at temperature 100° F., operates on a cycle composed of an isothermal expansion, an adiabatic expansion, an isothermal compression, and an isovolumic. Is this a reversible cycle? Why?

6. Draw cycle described in 5 above and determine:

(1) Heat supply (positive or negative) during each process.

(2) Work done (positive or negative) during each process.

(3) Efficiency of cycle.

(4) Carnot efficiency with same temperature limits; $\gamma = 1.41$; $R = 53.3$; $W = \frac{1}{2}$ lb.; $r = 2$ for the isothermal expansion.

7. One-half of a pound of air is enclosed in a cylinder fitted with a moveable piston. It occupies a volume of 3 cu. ft., exerts a pressure of 100 lbs. per sq. in., and the area of the piston is 1 sq. ft. The gas is expanded at constant pressure to a volume of 6 cu. ft.; the pressure is then dropped at constant volume to a value of 15 lbs. per sq. in.; the gas is then compressed at constant pressure to a volume of 3 cu. ft.; lastly the pressure is raised to 100 lbs. per sq. in. at constant volume.

(*a*) Draw the cycle to PV coördinates and indicate values of pressure, volume and temperature at the four corners.

(*b*) Find the net work done by the gas during one cycle.

(*c*) Find the heat supplied or rejected during each process and the net heat change.

(*d*) Find the efficiency of the cycle.

(*e*) Assuming the use of one hot and one cold body is this cycle reversible? Why?

8. (*a*) Draw the Carnot cycle to PV and Tϕ coördinates for the following conditions. Two pounds of nitrogen are used as working substance. The maximum temperature is 1500° F. and the maximum pressure is 200 lbs. per sq. in. The ratio of isothermal expansion is 2. The minimum temperature is 50° F.

(*b*) Find the heat supplied, the heat rejected and the work done.

(*c*) Find the efficiency of the cycle, or of an engine using the cycle.

9. (*a*) Determine the efficiencies of Carnot cycle engines using gaseous working substances when the hot body and the cold body have the following temperatures respectively:

Hot Body Temperature	Cold Body Temperature
(1) 3000° F.	500° F.
(2) 1500° F.	0° F.
(3) 1500° F.	500° F.

(*b*) Which is the more effective method of increasing the efficiency, raising T_1 or lowering T_2? Why?

10. If the maximum and minimum temperatures and pressures are both given, what must be the ratio of isothermal expansion in a Carnot cycle engine using gas as a working substance?

11. An engine operating on the Joule cycle has the following conditions at the end of compression $V_a = 0.5$ cu. ft.; $p_a = 70$ lbs. per sq. in.; $T_a = 1800°$ abs. After constant pressure expansion the volume is 0.75 cu. ft. For the gas used $C_p = 0.26$ and $C_v = 0.19$. Temperature at end of expansion is 900° abs. and at the beginning of compression is 600° abs. Find

(1) Net work of cycle.

(2) Efficiency of cycle.

12. A gas engine operating on the Otto cycle uses 0.15 lb. of gas per cycle having $C_p = 0.2056$, and $C_v = 0.1457$. The pressure at end of compression is 75 lb. per sq. in. and the temperature is 1000° abs. At end of explosion line the temperature has risen to 2500° abs.; at the end of expansion the temperature is 1800° abs. and at the beginning of compression it is 720° abs. Find:

(1) Work during expansion and compression.

(2) The heat sent into, and the heat sent out of, the system.

(3) The efficiency of the cycle.

13. A gas engine operating on the Diesel cycle uses an oil, the products of combustion of which have a gamma value of 1.41. The weight of gas used is 0.15 lbs. The temperature at end of compression is 1800° abs. and at end of constant pressure expansion is 2000° abs. The clearance is 8 per cent and the piston displacement is 2 cu. ft. ($C_p = 0.2056$, $C_v = 0.1457$.) Find:

(*a*) Heat added.

(*b*) Heat rejected.

(*c*) The pressure obtained at beginning and end of each process.

(*d*) The entropy changes for each line.

14. A Rider hot air engine, operating on the Stirling cycle, uses 0.066 lbs. of air. The temperature of the hot body is 2000° abs. and that of the cold body is 600° abs., the initial volume being 0.8 cu. ft. and the final volume being 1 cu. ft. ($C_p = 0.237$, $C_v = 0.169$.) Lowest pressure in cycle = 14.7 lbs. per sq. in. Find:

(*a*) The pressures at the beginning and end of expansion, and at the end of compression.

(*b*) Net work of cycle. (*c*) Efficiency of cycle.

15. A Diesel gas engine operates with a gas having $C_p = 0.22$ and $C_v = 0.156$. The pressure at the end of the compression is 550 lbs. per sq. in.; the volume is 0.1 cu. ft. and the temperature 2000° abs. At the beginning of the adiabatic expansion the temperature is 2400° abs. and at the end of same it is 1200° abs. Find all necessary temperatures, pressures and volumes to determine:

(*a*) The work done during the constant pressure and adiabatic expansions.

(*b*) Work during adiabatic compression.

(*c*) Efficiency of cycle.

16. A gas engine working on the Otto cycle has a pressure at the beginning of compression = 13 lbs. per sq. in. Find the clearance to give a compression of 91 lbs. per sq. in., assuming the exponent for compression to be 1.22. If the initial temperature is 60° F. find the temperature at the end of compression.

17. Suppose a gas engine is working on the Otto cycle, under the following conditions: Compression pressure = 80 lbs. per sq. in.; pressure after explosion = 240 lbs. per sq. in. The volume at the beginning of compression is 6 cu. ft., and at the end of compression it is 2 cu. ft. ($\gamma = 1.41$.) Find work done for each step of cycle and the net work done.

18. With the same data as in the previous problem, find the efficiency of the cycle, the heat supplied per cycle and the heat rejected.

19. If a gas engine working on the Otto cycle has 25 per cent clearance, find the efficiency of the cycle. ($\gamma = 1.41$.)

20. Find the horse power of a gas engine making 100 cycles per minute, each cycle transforming 50 B.t.u. into work.

21. How many B.t.u. per minute will be required to develop 100 H.P. in a single-acting 4-stroke cycle gas engine running 200 R.P.M. if the efficiency of the cycle is 30 per cent? What would be the heat input per cycle?

CHAPTER IX.

1. If liquid with a constant specific heat equal to 0.5 vaporizes at a temperature of 150° F. when the pressure is atmospheric and solidifies at 2° F.; (*a*) What is the value of q for atmospheric pressure figured above the temperature of fusion? (*b*) What is the value of q figured above 32° F. as a datum?

2. A certain liquid has a variable specific heat given by the following equation, $C_p = 0.7 + 0.003\,t - 0.0001\,t^2$ in which t stands for temperature in Fahrenheit degrees. It vaporizes at a temperature of 100° F. under atmospheric pressure and solidifies at $-35°$ F. What is the numerical value of the heat of the liquid figured above temperature of fusion as a datum?

3. When one pound of a certain liquid vaporizes under a pressure of 35 lbs. per sq. in., there is a volume change of 50 cu. ft. The total amount of heat added to cause vaporization of liquid already at vaporizing temperature is 900 B.t.u. (*a*) What is the numerical value of the latent heat of vaporization? (*b*) What is the numerical value of the external latent heat of vaporization? (*c*) What is the numerical value of the internal latent heat of vaporization?

4. A certain liquid has a constant specific heat of 0.85; it solidifies at $-25°$ F. and vaporizes under a pressure of 50 lbs. per sq. in. at a temperature of 175° F. The total heat required to raise the temperature of one pound of material from 60° F. and so cause total vaporization at a pressure of 50 lbs. per sq. in. is equal to 1250 B.t.u. (*a*) What is the numerical value of q for these

conditions, figured above fusion temperature as a datum? (*b*) What is the numerical value of r for these conditions? (*c*) What is the numerical value of λ for these conditions?

5. The pressure of a certain saturated vapor at a temperature of 250° F. is 75 lbs. per sq. in. What will be the temperature of vaporization of this material under a pressure of 75 lbs. per sq. in.?

6. The latent heat of vaporization of a certain material under certain conditions is 525 B.t.u. and the heat of the liquid is 210 B.t.u. What will be the heat above the chosen datum associated with 5 pounds of the vapor of this material when it has a quality of 85 per cent?

7. Three-quarters of a pound of liquid and one-quarter of a pound of the vapor of that liquid have been standing in a closed vessel for a considerable length of time. The temperature of the liquid is 75° F. (*a*) What is the condition of the vapor? Why? (*b*) What is the temperature of the vapor? Why?

8. The average constant pressure specific heat of a certain vapor over a temperature range of 100 degrees starting at the temperature of vaporization at a pressure of 100 lbs. per sq. in. is 0.4. The internal latent heat of vaporization at this pressure is 700 B.t.u. The external work done during vaporization is 46,600 ft.-lbs. The specific heat of the liquid is constant and equal to 0.9 over the temperature range of 200 degrees between datum temperature and the temperature of vaporization under the pressure given above. What is the total heat associated with 10 lbs. of this superheated vapor at a temperature of 100 degrees higher than its temperature of vaporization at the assumed pressure?

9. A certain material has a latent heat of vaporization of 700 B.t.u. at atmospheric pressure. The constant pressure specific heat of its superheated vapor at atmospheric pressure is 0.5. If 7 lbs. of 100° superheated vapor are brought into intimate contact with 5 lbs. of 75 per cent vapor of the same material and the combination is maintained at atmospheric pressure, what will be the ultimate condition of the resulting product?

10. A gas occupying a vessel with internal volume of 1 cu. ft. exerts a pressure of 10 lbs. per sq. in. Another gas in a similar vessel exerts a pressure of 15 lbs. per sq. in. What will be the pressure on the walls if both gases simultaneously occupy one of the vessels?

11. A certain space is saturated with a vapor at a temperature of 250° F. which exerts a pressure of 17 lbs. per sq. in. Air at the same temperature as the vapor is pumped into the same space until the pressure has risen to 25 lbs. sq. in. What weight of air occupies each cubic foot of the space if the specific volume of air at 32° F. and 14.7 lbs. per sq. in. is 12.387?

12. A certain liquid is vaporized in a closed vessel from which the vapor is allowed to escape as fast as generated. The pressure of the vapor in question at the temperature within the vessel should be 100 lbs. per sq. in. but a pressure gauge indicates a pressure of 102 lbs. per sq. in. The excess is supposed to be due to air mixed with the vapor. The value of R for air is 53.34. How much air must be present per cubic foot of space? Assume temperature of 100°.

CHAPTER X.

1. By means of the steam table find the temperature, total heat, heat of the liquid, internal latent heat, and external latent heat for one pound of dry saturated steam having the following absolute pressures in lbs. per sq. in.: 15, 50, 80, 125, 175 and 300.

2. Determine from the steam table the space filled by 10 lbs. of dry saturated steam under an absolute pressure of 200, 175, 135, 100, 80 and 10 lbs. per sq. in.

3. By means of the steam table determine the number of pounds of steam that will be required to fill 10 cu. ft. when under an absolute pressure of 200, 115, 40 and 5 lbs. per sq. in.: (*a*) If the quality is 100 per cent; (*b*) If the quality is 80 per cent.

4. Determine by means of the steam table the entropy of the liquid, and

the entropy of vaporization for 1 lb. of steam under the following absolute pressures: 15, 50, 80, 125, 175, and 300 lbs. per sq. in. (*a*) If the quality is 100 per cent. (*b*) If the quality is 90 per cent.

5. Assuming the specific heat of water to be unity, compute the entropy of the liquid for a pound of water heated to the point of vaporization for an absolute pressure of 100 lbs. per sq. in. Find the per cent error by comparison with steam table values.

6. Compute the entropy of vaporization for a pound of steam having the following temperatures: 400, 300 and 200° F., the corresponding latent heats being 827.2, 909.5 and 977.8 B.t.u.

7. (*a*) Determine from the steam tables the amount of heat required to make a pound of steam, having a pressure of 100 lbs. per sq. in. abs., by heating the water to the temperature of vaporization from a temperature of 100° F., and then vaporizing until the quality is 80, 90 and 100 per cent.

(*b*) Draw the Tϕ-diagram for (*a*).

8. With the same data as in (7) find:

(*a*) The amount of heat required to superheat the steam 100°, 200°. Take $C_p = 0.50$. (*b*) Draw the Tϕ-chart for (*a*).

9. Determine by means of the steam table the quality of steam containing 1029 B.t.u. of intrinsic heat and having a pressure of 125 lbs. per sq. in. abs.

10. Determine by means of the steam table the intrinsic heat added to 1 lb. of water in heating it from 90° F. to 444° F., if the pressure is kept constant at 120 lbs. per sq. in. abs.

11. Heat is added to 1 lb. of steam at a constant pressure of 80 lbs. per sq. in. abs., thereby increasing the quality from 0.4 to 0.9. Find by means of steam tables: (*a*) How much heat is added. (*b*) How much internal heat is added. (*c*) How much external heat is added. (*d*) How much work in ft. lbs. is done. (*e*) How much the volume of the steam is increased.

12. Find by means of the steam table the heat required to evaporate 50 lbs. of water having a temperature of 102° F. when pumped into a boiler, the steam pressure in which is 125 lbs. per sq. in. abs.

13. If the temperature of the feed water is 200°, find by means of the steam table the amount of heat required to make 100 lbs. of steam having a quality of 95 per cent, the pressure being constant at 175 lbs. per sq. in. abs.

14. By means of the steam table determine the volume occupied by the steam in the previous problem.

15. Compute the external work performed in vaporizing 4 lbs. of water into dry steam under a constant pressure of 100 lbs. per sq. in. abs. Compare with tabular value.

16. Find the pressure equivalent to the internal force which must be overcome in vaporizing 1 lb. of water under a pressure of 100 lbs. per sq. in. abs.

17. Steam in a boiler is under an absolute pressure of 100 lbs. per sq. in., and is superheated 150.6°.

(*a*) Find by Tumlirz equation the specific volume of this steam.

(*b*) Find the total heat of this steam if its specific heat is 0.51.

(*c*) Show part (*b*) on a Tϕ-chart.

18. Find the heat of superheat, and the total intrinsic heat energy for the steam in the previous problem.

19. Sketch on a Tϕ-chart the water curve and saturation curve for a pound of water vapor between the pressure limits of 2 and 200 lbs. per sq. in. abs. showing the intermediate values for the following pressures, 10, 50 and 100.

(*a*) Mark on the sketch the value of the temperature and the entropy for each point, estimating the distances as closely as possible.

(*b*) Give the numerical values for, and show what areas represent, the heat of the liquid, and the latent heat for the two limiting pressures given above.

(*c*) If you had drawn the above sketch to the following scale: 1″ = 100° of temperature and 1″ = $\frac{1}{2}$ unit of entropy, how many sq. in. would represent the total heat of the steam for the pressure of 200 lbs. per sq. in. abs.? Indicate this area on your sketch.

20. Find the volume of a boiler containing 1000 lbs. of water and 6 lbs. of

dry saturated steam under a pressure of 135 lbs. per sq. in. abs. What per cent of this volume is occupied by each? What per cent of the total heat energy above 32° is contained in each?

21. How much water having a temperature of 60° F. will be required to condense 3000 lbs. of steam per hour, the quality of the steam being 90 per cent and the pressure within the condenser being 1 lb. per sq. in. abs.? How much, if the condenser pressure is 5 lbs. per sq. in.? Assume the cold water to be thoroughly mixed with the steam.

22. Four pounds of steam having an absolute pressure of 125 lbs. per sq. in. are condensed by flowing into 200 lbs. of water, the temperature of which is thereby raised from 60 to 80° F. Find the initial quality of the steam.

23. Find the total heat necessary to change 500 lbs. of water from a feed pump temperature of 80° F. into superheated steam having a temperature of 1000° F., the boiler pressure remaining constant at 125 lbs. per sq. in. abs. Take the specific heat of superheated steam for this range to be 0.50.

24. One pound of water at a temperature of 60° F. enters a boiler and is vaporized under a pressure of 115 lbs. per sq. in. abs. until it becomes dry and saturated steam. How many pounds of water might have been vaporized if the same amount of heat had been added to the water at a temperature of 212° and changed into dry steam at this temperature?

25. How much steam with a quality of 90 per cent and an absolute pressure of 3.00 lbs. per sq. in. is condensed in a surface condenser using 14,500 lbs. of circulating (or condensing) water per hour, the water entering the condenser at 55° F. and leaving it at 115° F.?

26. An engine develops 15 horse power using 28 lbs. of steam per horse-power hour. If the engine exhausts its steam with a quality of 85 per cent and pressure of 15 lbs. per sq. in. abs. to the atmosphere, find the amount above 32° F. discharged per minute. What would be the maximum amount of heat that could be abstracted per minute from this exhaust steam by means of water which is to attain a temperature of 200°?

27. For each pound of coal having a heating value of 14,300 B.t.u., a certain boiler evaporated 10 lbs. of water into steam, with quality of 95 per cent, the pressure being constant at 145 lbs. per sq. in. abs., and the temperature of the feed water being 188° F.

Suppose that another boiler, using this same kind of coal, evaporated 9 lbs. of water into dry steam, under constant pressure of 125 lbs. per sq. in., from a feed water temperature of 121° F.

Find the heat given water and steam by each boiler, and the boiler efficiencies (efficiency = heat supplied water and steam ÷ heat supplied boiler).

28. Determine by means of the Tϕ-diagram the pressure at which steam will become dry and saturated by expanding reversibly and adiabatically from a pressure of 100 lbs. per sq. in. and a superheat of 50° F.

29. Determine by means of the Tϕ-diagram the final condition of steam which expands reversibly and adiabatically from an initial superheat of 150° F. at a pressure of 150 lbs. per sq. in. to a final pressure of 15 lbs. per sq. in.

30. Determine by means of the Tϕ-diagram the volume of 10 lbs. of 80 per cent steam at a pressure of 75 lbs. per sq. in.

31. By means of the values given in the steam table draw a Tϕ-diagram for steam to such a scale as to show the extreme upper parts of the water and saturation lines. Explain the shape produced.

32. Draw by means of values given in the steam tables a physical equilibrium diagram for liquid water and water vapor between the critical temperature and 100° F. similar to part of Fig. 34. (Use care in choosing scales so that entire diagram can be drawn on sheet chosen.)

33. Determine by means of the Mollier Chart the final conditions of steam expanding reversibly and adiabatically from an initial superheat of 400 degrees at a pressure of 100 lbs. per sq. in. to a final pressure of 10 lbs. per sq. in.

34. Find from the Mollier Chart the heat which must be supplied to change 10 lbs. of 95 per cent steam at 20 lbs. per sq. in. to steam superheated 200 degrees F. at the same pressure.*

* The Ellenwood Chart may be used instead of the Mollier in this and subsequent problems.

CHAPTER XI.

1. One pound of dry saturated steam at an absolute pressure of 100 lbs. per sq. in. is expanded along a reversible adiabatic until its pressure is 50 lbs. per sq. in. abs. Find its new quality, volume and the work done.

2. Find the same quantities with the same data as in the previous problem, except the second pressure, which is 15 lbs. per sq. in.

3. Steam is formed in a boiler at a constant pressure of 100 lb. per sq. in. abs. and with such a quality that a pound occupies only 3.54 cu. ft. Heat is added at constant pressure until the steam is superheated 150 degrees. (*a*) Determine the quality at the beginning of the superheating process. (*b*) Find the total heat added and the heat of superheat. (*c*) Show on Tϕ-chart.

4. How much external work was done in the previous problem during the period of superheating? What is the total intrinsic heat energy at the point of maximum superheat?

5. Steam is contained in a cylinder, at an absolute pressure of 125 lbs. per sq. in. and a quality of 80 per cent. It expands isentropically to 25 lbs. per sq. in. abs. (*a*) Find volume occupied by steam before expansion. (*b*) Find quality at end of expansion. (*c*) Find work done in ft.-lbs. during adiabatic expansion.

6. For previous problem find the total heat of steam at the beginning and at the end of expansion; also the intrinsic heat energy used up in expanding adiabatically.

7. Steam with a superheat of 200 degrees and an absolute pressure of 125 lbs. per sq. in. expands at constant entropy in a cylinder, until its pressure has been reduced 80 per cent. How much work has been accomplished?

(*a*) What is the total heat of the steam at point of maximum superheat?

(*b*) Show on PV- and Tϕ-charts.

8. If the steam of previous problem expands further to 15 lbs. per sq. in., find quality and intrinsic heat at end of the expansion.

9. Steam in a cylinder at a pressure of 25 lbs. per sq. in. abs. and a quality of 90 per cent is compressed along a reversible adiabatic until just dry and saturated.

(*a*) What are the temperature, pressure, and specific volume at end of compression?

10. Steam at an absolute pressure of 175 lbs. per sq. in. is confined in a metal cylinder. Heat is abstracted at constant volume until its temperature becomes 240° F. The steam is initially dry and saturated.

(*a*) Find the quality at the end of change.

(*b*) Find the amount of heat abstracted during the process.

(*c*) Show on PV- and Tϕ-fields.

11. Superheated steam exists in a cylinder at a pressure of 25 lbs. per sq. in. abs. and a temperature of 366° F.

(*a*) Determine volume of steam at this point.

(*b*) Isothermal compression occurs until the steam is just dry and saturated; how much heat has been abstracted? Show on Tϕ-chart.

12. Steam at a pressure of 125 lbs. per sq. in. abs. and a quality of 80.4 per cent has heat abstracted at constant volume until the pressure becomes 15 lbs. per sq. in. abs.

(*a*) Find quality at end of the process.

(*b*) Find volume at beginning and end of process.

(*c*) Find intrinsic heat change during process.

13. One pound of steam under a pressure of 120 lbs. per sq. in. abs. expands isentropically until its pressure is reduced 50 per cent; find its new quality, if its initial quality was 0.2, 0.4, 0.6, 0.8, 1.0. Show each of the above expansions on Tϕ-chart. Tabulate results.

14. If a pound of dry saturated steam under a pressure of 150 lbs. per sq. in. abs. is cooled at constant volume to a temperature of 59° F., find its new quality, pressure and the heat removed. Show on Tϕ-chart.

15. Steam with 70 per cent quality is heated at a constant pressure of 125

lbs. per sq. in. abs. until it becomes just dry and saturated. More heat is then added isothermally until a pressure of 25 lbs. per sq. in. abs. is reached.

(*a*) Find heat added during both processes.

(*b*) Find intrinsic heat gained during each operation.

16. If a pound of steam with quality of 70 per cent and pressure of 20 lbs. per sq. in. abs. has heat removed from it while maintained at constant volume, thereby reducing its pressure to 15 lbs. per sq. in. abs., find its final quality and the heat removed.

17. Dry saturated steam having a pressure of 165 lbs. per sq. in. abs. has heat abstracted from it while maintained at constant volume until its pressure becomes 25 lbs. per sq. in abs.

(*a*) Find the heat abstracted.

(*b*) Find the final quality of the steam.

(*c*) Find intrinsic heat change during process.

18. Fifteen pounds of dry saturated steam under a pressure of 110 lbs./sq. in. abs. are enclosed in a metallic tank, and this tank is then immersed in 1000 lbs. of water at a temperature of 60° F. If the tank is left in the cold water until the pressure within has reached atmospheric (14 7), find the temperature of the cold water and the quality of the steam. (Assume the tank to be a perfect conductor and that there are no heat losses.)

19. Given a pound of steam with a quality of 80 per cent and a pressure of 100 lbs. per sq. in. abs. Find the heat required and the work done to double its volume at constant pressure.

20. Suppose a pound of dry saturated steam having an absolute pressure of 150 lbs. per sq. in. expands isothermally until its volume is doubled; find the work done, heat supplied and final pressure.

21. Work previous problem, starting with the steam having a quality of 45 per cent.

22. Find the work done in ft.-lbs. in superheating one pound of steam 100 degrees at a constant pressure of 68 lbs. per sq. in. abs. Find the change in the intrinsic energy of the steam. Find the change in volume and the external work.

23. Suppose one pound of steam under an abs. pressure of 100 lbs. 1 sq. in. and a quality of 90 per cent expands along reversible adiabatic to a pressure of 2 lbs./sq. in. abs. Find its new quality, volume and the work done.

24. Suppose the steam in the previous problem to have been superheated 200 degrees; find the same quantities.

25. Given a pound of steam under a pressure of 15 lbs./sq. in. abs. and quality of 50 per cent. If heat is added while it is maintained at constant volume until its quality is unity, find the heat required, the final temperature and the pressure.

CHAPTER XII.

1. An engine using water vapor and operating on the Carnot cycle, as in Sect. 91, between the pressure limits of 185 lbs. and 25 lbs. per sq. in. begins adiabatic expansion at 80 per cent quality.

(*a*) Find the qualities at end of expansion and at beginning of adiabatic compression.

(*b*) Find the work accomplished in ft.-lbs.

(*c*) Find the efficiency of the cycle by two methods.

(*d*) Show the cycle on both PV- and Tϕ-fields.

2. With the same initial temperature and the same back pressure as in previous problem, suppose adiabatic expansion to start from a point in the superheated region where the pressure is 60 lbs. per sq. in. $C_p = 0.53$.

(*a*) Determine conditions of the steam at the end of the adiabatic expansion.

(*b*) Find the work accomplished in ft.-lbs.

(*c*) Find the efficiency of the cycle.

(*d*) Show the cycle on both PV- and Tϕ-fields.

3. Given the initial pressure of a Carnot Cycle with steam as working substance at 135 lbs. per sq. in. and the final pressure 25 lbs. per sq. in. The

isothermal expansion starts with water and the steam at the beginning of adiabatic expansion is just dry and saturated.

(*a*) Find the heat added and the heat rejected.

(*b*) Find the intrinsic energy at the beginning and end of each line of the cycle.

(*c*) Find the efficiency of the cycle.

(*d*) Show on Tϕ-chart.

4. What is the efficiency of the Carnot cycle for a steam engine receiving dry saturated steam at a pressure of 150 lbs. per sq. in. abs. and exhausting it at a pressure of 15 lbs. per sq. in. abs.

Show on the PV- and Tϕ-charts.

5. If the exhaust pressure in the previous problem had been 2 lbs. per sq. in. abs. what would have been the efficiency?

Show on PV- and Tϕ-charts.

6. If the engine in problem 4 exhausts 8000 B.t.u. per min., what is its power?

7. If the engine in problem 5 exhausts 8000 B.t.u. per min., what is its power?

8. The adiabatic expansion of a Carnot cycle which starts with liquid at the temperature of vaporization begins with a steam pressure of 25 lbs. per sq. in. abs. and a temperature of 344.4° F.

This expansion ends when a temperature of 213° F. has been reached. Find the heat supplied, heat rejected and the efficiency of the cycle. Check the efficiency by a second method.

Show on Tϕ-chart.

9. If the cycle of the previous problem were reversed, and we were to use some suitable working substance, what would be the coefficient of performance for a cooling machine? For a warming machine?

(Coef. of perf. = result ÷ expenditure.)

Show on the Tϕ-chart.

10. An engine working on the Carnot cycle transforms 113.94 B.t.u. into work per cycle. If the temperature of the steam initially is 358.5° F. and finally 250.3° F., find the initial state of the steam. Show on the Tϕ-chart.

11. If a steam engine, working on the Carnot cycle with an efficiency of 20 per cent requires 3000 lbs. of dry saturated steam per hour the pressure being 100 lbs. per sq. in. abs., find the exhaust pressure. How much heat does the condenser remove per hour? What is the horse power of this ideal engine? Show on the Tϕ-chart.

12. If a boiler supplies dry saturated steam with a pressure of 150 lbs. per sq. in. abs. to an engine operating on the Carnot cycle, in which the back pressure is 5 lbs. per sq. in. abs., find the work done, the heat flow for each path, the net work of the cycle, and its efficiency. Show on Tϕ- and PV-charts. Check your net work by another method.

13. Solve previous problem starting with a quality of 90 per cent.

14. An engine operates on the Clausius cycle between the pressure limits of 185 lbs. per sq. in. and 25 lbs. per sq. in. Adiabatic expansion occurs from a temperature of 575° F. $C_p = 0.54$. (*a*) Determine the total heat added ΔQ_1. (*b*) Determine the total heat rejected ΔQ_2. (*c*) Determine the work accomplished during adiabatic expansion. (*d*) Find the efficiency of the cycle.

15. Suppose the engine of the previous problem to operate on the Rankine cycle with release at 25 lbs. per sq. in. and a back pressure of 5 lbs. per sq. in. abs. (*a*) Determine the efficiency of this cycle. (*b*) What would be the efficiency of a Carnot cycle operating between the same temp. limits.

16. Find the difference in the heat supplied, heat rejected, work done and the efficiency of a Carnot and Clausius cycle when each engine receives dry saturated steam having an abs. pressure of 110 lbs. per sq. in., the back pressure being 20 lbs. per sq. in. abs.

Show on the PV- and Tϕ-charts.

17. Find the heat supplied, heat rejected, work done and the efficiency of an engine working on the Clausius cycle receiving steam with a pressure of 140 lbs. per sq. in. abs. and 150° of superheat, if the exhaust pressure is 15 lbs. per sq. in. abs.

Show the PV- and Tϕ-charts.

18. Find the heat supplied, heat rejected, work done and efficiency of an engine working on the Rankine cycle in which the initial steam pressure is 130 lbs. per sq. in. abs. with 150° of superheat; and the temperature at the end of adiabatic expansion is 240.1° F. while the exhaust pressure is 15 lbs. per sq. in. abs.

Show on PV- and Tϕ-charts.*

19. Given a feed pump cycle (rectangular PV-diagram) working between the pressure limits of 135 lbs. per sq. in. and 15 lbs. per sq. in. The steam is superheated 100° F. ($C_p = 0.55$.) (*a*) Find the volume of the cylinder required. (*b*) Find the efficiency of the cycle. (*c*) Show on PV- and Tϕ-charts.

20. Find the heat supplied, heat rejected and work done per pound and the efficiency of a feed pump cycle in which the heat abstraction begins when the steam has a pressure of 140 lbs. per sq. in. abs., and 50° of superheat. The constant volume change ends when a temperature of 213° F. has been reached.

21. Suppose an engine working on the Rankine cycle receives dry saturated steam having a pressure of 200 lbs. per sq. in. abs., then expands it until its pressure drops 90 per cent. The condenser pressure is 4 lbs. per sq. in. abs.

(*a*) Find the efficiency of the cycle.

(*b*) If the engine requires $\frac{1}{4}$ of a pound of steam per cycle find the volume of the cylinder necessary for this ideal cycle.

(*c*) With the same data as in (*b*) find the horse power if the engine runs 300 cycles per min.

(*d*) Plot the PV- and Tϕ-charts for this cycle estimating distances as closely as possible.

(*e*) What is the horse power per cu. ft. of piston displacement?

22. With the same initial steam and the same back pressure as in previous problem, find the same quantities for the Clausius cycle.

23. With the same initial steam and the same back pressure as in 21 find the same quantities for the cycle in which there is no expansion of the steam.

CHAPTER XIII.

1. How much heat is transformed into work by an engine delivering 100 h.p. for 24 hrs.?

2. Find the i.h.p. of a 10-in. by 12-in. engine, double acting, running 200 rev. per min. when the m.e.p. from the indicator diagram is 30 lbs. per sq. in.

3. Suppose the engine of the previous problem had given an indicator diagram, having an area of 2.7 sq. in., and a length of 3 in. If the scale of the spring used in the indicator had been 50 lbs. per in., find the i.h.p.

4. An engine 18 in. by 24 in., double acting, running 150 rev. per min. delivers 100 b.h.p. If the indicator diagram gives a m.e.p. of 25 lbs. per sq. in., find its mechanical efficiency, and the friction horse power.

5. Suppose the same engine, running at the same speed as in the previous problem, has its back pressure reduced 12 lbs. per sq. in. by means of a condenser thereby increasing the m.e.p. to 36 lbs. per sq. in. Find its i.h.p., b.h.p. and mechanical efficiency, assuming the same friction horse power as in the previous problem.

The addition of a condenser to this engine increased its power output by what per cent?

6. A certain engine working on the Rankine cycle uses 20 lbs. of steam per i.h.p. hr. If the steam has an abs. pressure of 150 lbs. per sq. in. and a temp. of 458.5° while the exhaust pressure is 4 lbs. per sq. in. abs., find the heat supplied per i.h.p. min. and the thermal efficiency on i.h.p.

7. If the engine of the previous problem has a mechanical efficiency of 90 per cent, what is the thermal efficiency on b.h.p.?

* The Ellenwood Chart may be used in probs. 18–23.

8. Suppose an engine in which the mechanical efficiency is 85 per cent requires 3000 lbs. of steam per hr. when delivering 100 h.p. If the steam has a pressure of 150 lbs. per sq. in. abs., and a quality of 98 per cent, find the heat required per b.h.p. min., and the thermal efficiency on the i.h.p. and on the b.h.p. The engine is working on the Rankine cycle, and has a back pressure of 2 lbs. per sq. in. abs.

9. With the same engine delivering the same power as in the previous problem, find the same quantities if the steam required is 2400 lbs. per hr., the steam having a pressure of 150 lbs. per sq. in. abs. and 100 degrees of superheat.

10. If a Diesel engine delivers 750 h.p. hrs. per bbl. of crude oil, find its thermal efficiency on b.h.p. A barrel of oil contains 336 lbs. and the calorific value of this oil is 18,500 B.t.u. per lb.

11. An $8'' \times 12''$ air compressor, while running 200 r.p.m. gives an indicator card having an area of 2.7 sq. in. for the head end, and 3 sq. in. for the crank end. The length of each card is 3 in. Scale of the spring is 60 lbs. per in. The piston rod is $1\frac{1}{2}$ in. in diameter. Find the i.h.p.

12. A Diesel engine uses 4 bbl. of crude oil per day of 24 hrs. (1 bbl. = = 336 lbs.) The heating value of the oil is 18,000 B.t.u. per lb. If the mechanical efficiency is 73 per cent and the thermal efficiency on b.h.p. is 28.3 per cent find the i.h.p., b.h.p. and oil used per b.h.p. hr. for this engine.

13. A 12,000-h.p. steam turbine requires 12.3 lbs. of steam per h.p. hr., when receiving steam having 100° of superheat and a pressure of 190 lbs. per sq. in. abs. The vacuum is 28 in., and the barometer stands at 30 in.

(*a*) Find the cycle efficiency for this turbine assuming it to operate on the Clausius cycle. (*b*) Find the delivered thermal efficiency. (*c*) What portion of the theoretical work of the cycle is actually delivered by the turbine?

14. With the same data as in the previous problem, find the actual amount of heat lost per min. in this turbine, in excess of that rejected by the ideal turbine operating under the same conditions.

15. A certain steam engine rated at 500 h.p. gives a total consumption curve which is a straight line. The total consumption at $\frac{1}{4}$ of rated load is 6250 lbs. of steam per hour while that at rated load is 14,000 lbs. per hour. Plot the total consumption and the water rate curves for this engine between zero and rated curves.

CHAPTER XIV.

1. Find the work done (in B.t.u.), the cycle efficiency and the water rate of an ideal turbine operating on the Clausius cycle with steam at a pressure of 130 lbs. per sq. in. abs. and 100° of superheat, while the back pressure is 15 lbs. per sq. in. abs.

2. Solve problem 1 using a back pressure of 1 lb. per sq. in. abs. By what per cent was the power of this ideal turbine increased by exhausting into a condenser?

3. Solve problem 1 using steam of the same pressure but having a quality of unity at the beginning of adiabatic expansion.

4. Discuss the results of the preceding problems, as to the effect of superheat and vacuum on the ideal water rates.

5. Find the net work of the cycle, the cycle efficiency, and the theoretical water rate of an ideal engine working on the Carnot cycle, for which the upper temperature is 344.4° and the lower is 193.22°. The pressure at the beginning of the adiabatic expansion is 25 lbs. per sq. in. abs.

6. Solve the previous problem, with all the conditions as above, except that the steam at the beginning of adiabatic expansion is dry and saturated at 344.4° F.

7. Given admission pressure = 130 lbs. per sq. in. abs., D = 100°, release press. = 25 lbs. per sq. in. abs., and back press. = 15 lbs. per sq. in. abs., find the theoretical work of the cycle, the cycle efficiency, and the theoretical water rate, of an engine operating on the Rankine cycle.

8. Solve problem 7 when the engine receives dry saturated steam.

9. Suppose an engine is working on the feed pump cycle, receiving steam with an abs. press. of 130 lbs. per sq. in. and 100° of superheat. Find the cycle efficiency, the theoretical water rate and the actual water per i.h.p. hr. if the indicated thermal efficiency is 3 per cent and back pressure is 15 lbs.

10. Solve problem 9, using dry saturated steam.

11. An engine is working on the Rankine cycle at a pressure of 130 lbs. per sq. in. abs. and superheated 100°; release occurs at a pressure of 25 lbs. per sq. in. abs.; back pressure is 15 lbs. per sq. in. abs. Find the theoretical water rate and the cycle efficiency. Find the actual steam used per b.h.p. hr. and the delivered thermal efficiency if this engine requires 30,000 lbs. of steam per day of 10 hrs. when delivering 100 h.p.

12. Given dry saturated steam having a temperature of 327.8°, and an exhaust temperature of 213°, find the net work of the cycle, the cycle efficiency, and the theoretical water rate of an engine working on the Carnot cycle.

13. Solve problem 12 for the Clausius cycle.

14. Solve problem 12 for the Rankine cycle, assuming release to occur when a temperature of 240.1° has been reached.

15. Solve problem 12 for the feed pump cycle.

16. Starting with the same $\triangle Q_1$ and working between the same temperature limits as in problem 12, find the net work of the cycle, the cycle efficiency and the theoretical water rate of an engine working on the Clausius cycle.

17. Solve problem 16 for the Rankine cycle, the release temperature being 240.1°.

18. Solve problem 16 for the feed pump cycle.

CHAPTER XV.

1. An engine has a piston displacement of 0.2 cu. ft. If its clearance is 10 per cent, and release takes place at 95 per cent of the stroke, find the weight of steam in the cylinder at release, the quality then being 70 per cent and the pressure 25 lbs. per sq. in. abs.

2. Find the quality of steam at cut off in a cylinder in which the piston displacement is 0.1278 cu. ft., clearance 10 per cent, cut off 25 per cent, steam pressure at cut off 115 lbs. per sq. in. abs., and the weight of steam in the cylinder at cut off 0.012 lbs.

3. Determine the quality or degree of superheat of the steam in an engine cylinder at cut off, its pressure then being 125 lbs. per sq. in. abs. and its weight 0.013 lbs. The piston displacement is 0.13 cu. ft., clearance 10 per cent, and the cut off takes place at 30 per cent of the stroke.

4. Find the weight of cushion steam in a $6'' \times 8''$ engine in which clearance is 15 per cent; compression begins at 15 per cent of the stroke; the back pressure is 14.7 lbs. per sq. in. abs., and the quality of the cushion steam at the beginning of compression is 95 per cent. Find the pressure and the quality at the end of this compression, assuming it to be adiabatic.

5. Suppose the compression in the previous problem is not adiabatic, but is such that the compression pressure is 30 lbs. per sq. in. abs., find the quality of the cushion steam at the end of the stroke.

6. An $8'' \times 10''$ engine running 300 r.p.m. double acting, with cut off taking place at 15 per cent of the stroke, and at a pressure of 120 lbs. per sq. in. abs., requires 35 lbs. of steam per i.h.p. hr. The compression begins at 40 per cent of the stroke with a quality of unity and a back pressure of 5 lbs. per sq. in. abs. Clearance = 10 per cent.

If this engine delivers 27 h.p. and has a mechanical efficiency of 90 per cent,

(*a*) Find the quality of steam at cut off.

(*b*) If the quality of the steam at the throttle is unity and "wire drawing" amounts to 5 pounds, find the delivered thermal efficiency.

7. For the previous problem, assume release to occur at 90 per cent of the stroke, with an abs. pressure of 30 lbs. per sq. in. What is the quality at this point?

8. Assuming the expansion line for problem 6 to follow the law, PV = const.

(*a*) Find the quality at 15, 20, 30, 40, 50, 65 and 90 per cent of the stroke. (Tabulate results.)

(*b*) Draw the quality curve for this expansion.

9. Suppose the engine in problem 6 had received steam with sufficient superheat to cause the quality at cut off to be 88 per cent, and that by means of a steam jacket the expansion line is made to follow the law, $PV^{.952} = \text{const.}$ Find the condition of the steam at release, which occurs at 90 per cent of the stroke.

10. The thermometer in a throttling calorimeter shows a temp. of 223°, the manometer reads 0.73 in. of mercury; and the barometer stands at 29.92. Find the quality of the steam entering the calorimeter if its pressure is 110.3 lbs. per sq. in. gauge.

11. Supposing that the thermometer in the above calorimeter had read 213°, all other readings being the same, what would be the quality? Discuss.

12. If by connecting to a condenser we may reduce the calorimeter pressure until the mercury manometer indicates 15.62 in. below atmosphere, find the quality of the steam below which the instrument could not be used with steam pressure and atmospheric pressure the same as in problem 10.

13. What pressure would you have to maintain in the calorimeter, in order to measure quality as low as 92 per cent, the steam pressure being 100 lbs. per sq. in. abs. and the degree of superheat in the calorimeter to be not less than 5°.

CHAPTER XVI.

Heck's formula for estimating cylinder condensation in cylinders which are not steam jacketed is

$$m = \frac{0.27}{\sqrt[3]{N}} \sqrt{\frac{S T_f}{pe}},$$

where m = fraction of moisture in the steam at any point "a" during expansion.

N = r.p.m.

$S = \dfrac{\text{nominal cyl. surface in sq. ft.}}{\text{nominal cyl. vol. in cu. ft.}}$

$= \dfrac{24}{l} + \dfrac{48}{d}$ { where d = diam. of cyl. in inches. and l = length of stroke in inches.

p = abs. press. in lb. per sq. in. at point "a."

$e = \dfrac{\text{total vol. up to point "a"}}{\text{piston displ.}}$.

T_f = a special temperature function of the pressure and is obtained from the following table by taking the difference between the values of K corresponding to the highest and the lowest pressures occurring in the engine.

p	K	p	K	p	K	p	K	p	K	p	K
1	175	15	210	50	269.5	90	321.5	160	389	230	441
2	179	20	220	55	277	100	332.5	170	397	240	447.5
3	183	25	229	60	284	110	343	180	405	250	454
4	186	30	238	65	291	120	353	190	413	260	460.5
6	191	35	246	70	297.5	130	362.5	200	420	270	467
8	196	40	254	75	304	140	371.5	210	427	280	473
10	200	45	262	80	310	150	380.5	220	434	290	479

Note.—For more complete table see page 112, vol. 1, "The Steam Engine," Heck.
This formula is most applicable over a range of cut-off from 16 to 65 per cent of the stroke.

1. Find the quality at cut off by Heck's formula for the intermediate cylinder of a triple expansion engine having a diam. of 10 in. and a stroke of 6 in., the speed being 402 r.p.m., cut-off press., 112.2; admission press., 130; and back press., 51 lbs. per sq. in. abs.; clearance, 10 per cent; cut off, 36.2 per cent. By test it was found that the actual quality at cut off was 85.7 per cent; find the per cent error by computing with the above formula.

2. A $17'' \times 24''$ engine running 72 r.p.m. gives a card which shows the adm. press. to be 133 lbs.; exhaust press., 16.2; cut off, 98.8 lbs. per sq. in. abs. Cut off occurs at 18.6 per cent stroke and clearance is 9.9 per cent. Find the quality at cut off by Heck's formula and find the per cent error if test shows the actual quality to be 72.9 per cent.

3. An $8'' \times 10''$ engine running 320 r.p.m. has cut off at 12 per cent with a pressure of 140; admission press. is 145, and back press. 4 lbs. per sq. in. abs.; clearance is 10 per cent. If a test shows the quality at cut off to be 59 per cent, find the per cent error in result obtained by Heck's formula.

4. A certain engine requires 18 lbs. of steam per i.h.p. hour when using dry saturated steam, but when the steam is superheated 340° it requires only 11 lbs. per i.h.p. hour.

The admission pressure is 150 lbs. per sq. in. abs. and the back pressure is 1 lb. in each case. Find:

(*a*) The per cent saving in steam required when superheat is used.

(*b*) The per cent saving in heat required when superheat is used.

(*c*) The per cent saving in steam required for ideal Clausius cycle when superheat is used.

(*d*) The per cent saving in heat required for ideal Clausius cycle when superheat is used.

(*e*) The difference in the saving in heat required for the actual and the ideal engine. This difference is due mainly to what?

5. An engine is supplied with dry saturated steam having an abs. pressure of 150 lbs. per sq. in.

Back pressure is 5 lbs. per sq. in. abs., the exhaust steam being liquified in a surface condenser.

When running without a steam jacket, the engine requires 20 lbs. of steam per i.h.p. hr.

When running with a jacket, it was found the engine required 18 lbs. per i.h.p. hr. for the cylinder, and 2 lbs. per i.h.p. for the jacket.

Supposing there is no loss of heat in returning the condensate from the jacket, or from the condenser to the boiler, and that the jacket pressure is maintained at 150 lbs. per sq. in. abs.,

(*a*) Find the heat required by the engine per i.h.p. hr. for each of the above cases.

(*b*) Find the ind. thermal efficiency for each of the above cases.

(*c*) Find the per cent saving in heat required per i.h.p. due to jacketing.

6. An engine running without a steam jacket requires 15.5 lbs. of steam per i.h.p. hr., and when running with the steam jacket requires a total of 14 lbs. per i.h.p. hr. The steam required by the jacket is 18 per cent of the total.

The admission press. and jacket press. are both 165 lbs. per sq. in. abs.; the steam admitted to them has a quality of unity.

In each case the condensate from the condenser is returned to the boiler at a temp. of 202° F., but the condensate from the steam jacket is delivered to the boiler at a temp. of 302°. Neglecting all leakage: (*a*) Find the heat supplied by the boiler per i.h.p. hr. for each of the above cases. (*b*) Find the per cent saving in heat required due to jacketing.

CHAPTER XIX.

1. (*a*) Lay out a symmetrical valve seat, with width of exhaust cavity 5 ins., of metal between exhaust cavity and port $1\frac{1}{2}$ ins., and of ports $1\frac{1}{4}$ ins. (*b*) Draw in its central position an external valve having steam and exhaust laps respectively $1\frac{1}{4}$ ins. and (negative) $\frac{1}{8}$ in. for the head end, and $1\frac{1}{8}$ ins. and

(positive) $\frac{1}{8}$ in. for the crank end; thickness of metal 1 in. Dimension and label completely.

2. Same as problem 1, except for an internal valve.

3. Construct a rectangular diagram of valve displacements for a valve having the dimensions given in problem 1, the angle of advance being $32\frac{1}{2}°$ and throw $2\frac{1}{2}$ ins., (*a*) for the H.E.; (*b*) for the C.E.

4. Same as problem 3, but for polar diagram of valve displacements.

5. With data of problems 1 and 3, construct the Sweet diagram for both ends of valve and show: (*a*) the crank and piston positions for all the events; (*b*) the angles of rotation of crank and eccentric for each of the periods; (*c*) the maximum openings to steam and exhaust; (*d*) the lead. (*e*) Construct an elliptical diagram from the Sweet diagram.

6. Same as problem 5, except for Zeuner diagram.

7. Same as problem 5, except for Bilgram diagram.

8. Given cut-off $\frac{3}{4}$ stroke, the amount of lead $\frac{1}{4}$ in., the maximum width of opening of the steam edge of the valve $1\frac{1}{4}$ ins., release 95 per cent of stroke for H.E. and 90 per cent for C.E. Determine, for both ends, the value of (*a*) the angle of advance, (*b*) throw, (*c*) steam lap, (*d*) exhaust lap, and (*e*) crank and piston positions for each event.

9. In problems 5 to 8, find the true positions of the piston in its stroke for each crank position found, the length of connecting rod being 6 times the length of crank. Let the eccentric circle represent the crank circle.

10. A swinging eccentric is pivoted diametrically opposite the crank at a distance of 8 ins. from the shaft center; the distance from pivot to eccentric center is $6\frac{7}{8}$ ins., the largest throw of the eccentric is $2\frac{3}{4}$ ins., the steam lap is $1\frac{1}{8}$ ins. and the exhaust lap is $\frac{1}{8}$ in. (*a*) Draw the path of the swinging eccentric as in Fig. 160. Locate three eccentric positions, and get the throws and angles of advance. (*b*) Draw the H.E. Bilgram diagram and show the corresponding positions of the steam and exhaust lap circles. (*c*) Determine the positions of the crank for all events.

11. Same as problem 10, but using the Sweet diagram.

12. Same as problem 10, but using the Zeuner diagram.

13. *Independent cut-off gear.* (*a*) Draw the Bilgram diagram for the main valve to give release 90 per cent and compression 85 per cent of stroke, lead $\frac{1}{8}$ in., and maximum steam opening $1\frac{1}{8}$ ins. (*b*) With same eccentric throw, and in position for 0, $\frac{1}{4}$ and $\frac{1}{2}$ cut-off, draw the lap circles for a cut-off valve having 1 in. negative lap and riding on an independent stationary seat. (*c*) Show the positions of the eccentrics with respect to crank (on H.E. dead center) for each cut-off. (*d*) Plot a diagram of openings similar to Fig. 171.

14. *Meyer Valve Gear.* (*a*) Data same as in problem 13 (*a*) for main valve. (*b*) With throw of cut-off eccentric 3 ins. and angle of advance 90°, construct the Bilgram diagram and draw the lap circles for the cut-off valve to give cut-off at 0, $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ strokes. (*c*) Show the positions of the eccentrics with respect to the crank (on H.E. dead center).

CHAPTER XX.

NOTE. — In the following problems it is assumed that the engine is double acting unless otherwise stated.

1. It is desired to build a simple engine to give 75 i.h.p. under the following conditions:

Cut off at 20 per cent.
Clearance = 10 per cent.
Steam press. = 140 lbs. per sq. in. abs.
Back press. = 2 lbs. per sq. in. abs.
R.P.M. = 200.

If the diagram factor for this type of engine is 0.9 find size of cylinder.

2. Suppose it is desired to build an engine to give 50 i.h.p., under the following conditions: Cut off 25 per cent; Clearance 12 per cent.; Steam

press. = 150 lbs. per sq. in. abs.; Piston speed = 400 ft. per min.; Back press. = 16 lbs. per sq. in abs. If the diagram factor for this type of engine is 85 per cent, find the diameter of the cylinder and select stroke and r.p.m.

3. Supposing that the engine in problem 1 has cut offs occuring from 10 to 50 per cent of stroke, find the i.h.p. for these extremes.

4. Given an engine $18'' \times 24''$ running 120 r.p.m. Back press. = 2 lbs. per sq. in. abs; Clearance = 10 per cent; Cut off = 40 per cent; Diagram factor = 85 per cent. Supposing the cut off to remain constant, find the i.h.p.'s corresponding to steam pressures of 50, 90 and 130 lbs. per sq. in. abs.

5. Find the weight of dry steam which must be supplied per i.h.p. hour for each case of problem 4, assuming the quality at cut off to be 80 per cent. Assume compression pressure to be 30 lbs. abs. and that the steam is dry and saturated at the end of compression.

6. A compound engine is to give 600 i.h.p. when using steam having an abs. press. of 150 lbs. per sq. in. and having a back press. of 2 lbs. per sq. in. abs. If the cylinder ratio is to be 4 and the total ratio of expansion 12, find the size of cylinders. The piston speed is to be 750 ft. per min. and the engine is to run 150 r.p.m. Take diagram factor as 0.8.

7. An $8'' \times 16'' \times 12''$ engine has cut off in the high press. cylinder at $\frac{1}{3}$ stroke, the admission pressure is 150 lbs. per sq. in. abs. and the back pressure is 2 lbs. per sq. in. abs. Assuming the expansion line to be hyperbolic, receiver drop and clearance to be zero, and no initial superheat, find: (*a*) The total ratio of expansion. (*b*) The receiver pressure. (*c*) The point of cut off in the low pressure cylinder. (*d*) The temperature range in each cylinder. (*e*) The portion of the work done by each cylinder. (*f*) The maximum force exerted on each piston rod.

Sketch the PV-diagram for this expansion marking the pressure at the end of the expansion in each cylinder, and the volume in terms of the volume at cut off in the high pressure cylinder.

8. Assume the cut off in the high pressure cylinder of the above engine is now made to occur at $\frac{1}{5}$ stroke, all other conditions remaining the same, find the same quantities as called for above.

9. Assume the above engine is now made to cut off at $\frac{1}{4}$ stroke in the high pressure cylinder and at $37\frac{1}{2}$ per cent stroke in the low, thus causing a receiver drop, find this drop and all the quantities called for above.

10. With the data of the previous problem find what portion of the total theoretical work was lost by the receiver drop.

11. With the result of the previous problem solve for the constant "K" used in equation 293 of the text. Then with this value of K find the total m.e.p. referred to the low pressure cylinder. Does this check results already obtained from problem 9?

CHAPTER XXII.

1. Determine the amount of work theoretically obtainable with a turbine per pound of steam admitted at a pressure of 150 lbs. per sq. in. and superheated 100 degrees and expanded to a 28-inch vacuum, (*a*) by calculation, (*b*) by means of the Tϕ-chart, and (*c*) by means of the Mollier Chart.*

2. Same as problem 1 except that the exhaust pressure is atmospheric.

3. Determine the amount of work theoretically obtainable with a turbine per pound of dry saturated steam at a pressure of 150 lbs. per sq. in. and exhausting into a 28-inch vacuum.

4. Same as problem 3, except that the exhaust pressure is atmospheric.

5. Plot a curve showing the variation in the amount of work obtainable per pound of steam used in a turbine with initial conditions varying from dry and saturated at 175 lbs. per sq. in. pressure to a superheat of 150 degrees above the temperature of saturation, the turbine exhausting into a 28-inch vacuum in every case.

6. Plot a curve showing the variation in the amount of work obtainable per pound of steam used in a turbine with initial conditions constant at 150

* (*d*) By the Ellenwood Chart

degrees superheat and 175 lbs. per sq. in. pressure but with exhaust conditions varying from atmospheric pressure to a vacuum of 29 inches.

7. Determine the theoretical water rate of a turbine operating under the conditions specified in problem 1.

8. Determine the theoretical water rate of a turbine operating under the conditions specified in problem 2.

9. Determine the theoretical water rate of a turbine operating under the conditions specified in problem 3.

10. Determine the theoretical water rate of a turbine operating under the conditions specified in problem 4.

11. The over-all efficiency of a certain turbo-generator unit is 0.67 when receiving steam at a pressure of 175 lbs. per sq. in. superheated 125 degrees and exhausting to a 28.5-inch vacuum. What is the water rate of the unit per k.w. hour?

12. If the generator efficiency for the unit in problem 11 be 94 per cent, what is the water rate of the turbine per d.h.p. hour?

13. Find the B.t.u. supplied per kw. minute for the unit described in problem 11. (On basis of ideal feed-water temperature.)

14. Find the thermal efficiency of the unit described in problem 11.

15. Assume a reciprocating engine to consume twenty pounds of steam per h.p. hour when exhausting at atmospheric pressure. How much work could theoretically be obtained from an exhaust steam turbine receiving this steam at a quality of 80 per cent and expanding to a 27.5-inch vacuum?

16. Find the theoretical velocity of the jet in an impulse turbine receiving steam at a pressure of 100 lbs. per sq. in. with a superheat of 100 degrees and expanding to atmospheric pressure.

17. Find the theoretical kinetic energy of the jet per pound of steam discharged in problem 16.

CHAPTER XXVI.

1. Plot a curve showing the variation of the theoretical efficiency of the Otto cycle due to changes in the compression ratio [(volume of piston displacement + clearance) ÷ clearance volume] from 3 to 10 and with $\gamma = 1.4$.

2. Plot a similar curve for $\gamma = 1.33$.

3. A certain internal combustion engine operating on natural gas uses 10.3 cu. ft. of gas per b.h.p. hour at rated load. The calorific value of the gas is 1050 B.t.u. per cu. ft. (*a*) How many B.t.u. are required per b.h.p. per hour at rated load? (*b*) What is the thermal efficiency at rated load? (*c*) What values of thermal efficiency would you expect at half and quarter loads? Give method used in arriving at answer.

4. A certain producer-gas plant delivers a b.h.p. per hour on one pound of coal, the calorific value of the coal being 13,500 B.t.u. per pound. What is the thermal efficiency of the plant?

CHAPTER XXVII.

1. The proximate analysis of coal as received is moisture, 7 per cent; volatile, 4 per cent; fixed carbon, 81 per cent; ash, 8 per cent. (*a*) What would be the proximate analysis on a basis of dry coal? (*b*) What would be the proximate analysis on a basis of dry combustible?

2. The proximate analysis of coal as received is moisture, 10 per cent; volatile, 30 per cent; fixed carbon, 50 per cent; ash, 10 per cent. (*a*) What would be the proximate analysis on the basis of dry coal? (*b*) What would be the proximate analysis on the basis of dry combustible?

3. The proximate analysis of a certain coal on the basis of dry coal is volatile, 20 per cent; fixed carbon, 70 per cent; ash, 10 per cent. What would be the proximate analysis of this fuel when containing 10 per cent moisture?

4. The proximate analysis of a certain coal on the basis of dry combustible is volatile matter, 34 per cent; fixed carbon, 66 per cent. (*a*) What would be the proximate analysis of this fuel on the basis of dry fuel when

containing 8 per cent ash? (*b*) What would be the proximate analysis of this fuel as received when containing 8 per cent ash and 12 per cent moisture?

5. Determine the probable ultimate analysis of an anthracite coal giving the following proximate analysis: volatile, 4 per cent; fixed carbon, 96 per cent.

6. Determine the probable ultimate analysis of an anthracite coal giving the following proximate analysis: volatile, 3.5 per cent; fixed carbon, 90 per cent; ash, 6.5 per cent. (On basis of *dry* coal.)

7. Determine the probable ultimate analysis of a bituminous coal giving the following proximate analysis: volatile, 25 per cent; fixed carbon, 75 per cent.

8. Determine the probable calorific values of the coal of problem 5 by means of Dulong's formulas.

9. Same as 8, but for the coal of problem 6.

10. Same as 8, but for the coal of problem 7.

11. Determine the probable higher calorific value of a petroleum distillate with a specific gravity indicated as 95 on the Baumé scale.

12. Same as 11, except for a specific gravity of 85 Baumé.

13. Same as 11, except for a specific gravity of 75 Baumé.

14. Plot a curve showing the probable variation of higher calorific values of petroleum products with specific gravities varying from 50 to 95 Baumé.

CHAPTER XXVIII.

1. What weight of oxygen will be required to burn 5 lbs. of carbon to carbon dioxide? What weight of air will be required? How much nitrogen will there be in this air?

2. What weight of oxygen will be required to burn $7\frac{1}{2}$ lbs. of carbon to carbon monoxide? What weight of air will be required? What weight of nitrogen will be contained in this air?

3. What weight of carbon dioxide will result from the combustion of 12 lbs of carbon? What would be the total weight of the products of combustion (all material present after combustion) if the carbon were burned with the theoretical air supply?

4. Seven pounds of carbon are burned with air to carbon monoxide. (*a*) What weight of carbon monoxide is formed? (*b*) What is the total weight of the products of combustion?

5. Fifteen pounds of carbon are burned in oxygen to carbon dioxide. (*a*) What weight of carbon dioxide results? (*b*) How much heat is liberated? (*c*) How much heat would have been liberated if the combustion had taken place in air instead of in oxygen?

6. Seventeen pounds of carbon are burned to carbon dioxide in an apparatus which makes it possible to complete the combustion in one second, and an equal quantity is burned to the dioxide in an apparatus which requires one hour to complete the combustion. Is there any difference in the amount of heat liberated in the two cases? Why?

7. Three pounds of carbon are burned in air to carbon dioxide. (*a*) What will be the weight of the products of combustion if twice the theoretical quantity of air is used? (*b*) What will be the quantity of heat liberated?

8. Four pounds of carbon are burned in air to carbon monoxide. (*a*) What will be the weight of the products of combustion? (*b*) What will be the quantity of heat liberated?

9. A quantity of carbon monoxide containing 3 lbs. of carbon is burned with theoretical air supply to carbon dioxide. (*a*) What will be the weight of the products? (*b*) What quantity of heat will be liberated?

10. Ten pounds of carbon monoxide are burned with $1\frac{1}{2}$ times the theoretical air supply. (*a*) What will be the weight of the products? (*b*) What quantity of heat will be liberated?

11. Twelve pounds of carbon are burned first to carbon monoxide with the theoretical quantity of air and then the resultant carbon monoxide is burned with $1\frac{1}{4}$ times the theoretical air. (*a*) What weight of carbon monoxide is formed by the first reaction? (*b*) What is the total weight of gas after the first reaction? (*c*) What quantity of heat is liberated during the first reaction? (*d*) What weight of carbon dioxide is formed by the second reaction? (*e*) What is the total weight of gas present after completion of second reaction if no gas is lost during either reaction nor between reactions? (*f*) What quantity of heat is liberated during the second combustion? (*g*) What is the total quantity of heat liberated as result of both reactions and how does this compare with what would have been obtained had the 12 lbs. of carbon been burned directly to carbon dioxide? (*h*) How does the weight of products obtained by using two reactions compare with the weight that would have been obtained by burning directly to the dioxide with $1\frac{1}{4}$ times theoretical air?

12. Eight pounds of carbon are burned with air containing sufficient oxygen to burn 7 lbs. of carbon to carbon dioxide. (*a*) What are the weights of the various products of combustion? (*b*) What is the percentage composition of the products of combustion on a volume basis? (*c*) What quantity of heat is liberated?

13. The analysis of the gases obtained by burning carbon in air shows 15 per cent by volume of carbon dioxide. (*a*) What is the excess coefficient? (*b*) How many pounds of air were used per pound of carbon burned?

14. The analysis of gases obtained by burning carbon in air gives 79 per cent nitrogen and 7 per cent oxygen. (*a*) What is the excess coefficient? (*b*) What weight of air was used per pound of carbon?

15. Five pounds of carbon are burned in air with an excess of 50 per cent, the combustion taking place at constant pressure. (*a*) What temperature rise will result? (*b*) What will be the final temperature if all material is at temperature of 70° F. before the start of the combustion?

16. Sixteen cubic feet of carbon monoxide are burned with the theoretical quantity of oxygen within a vessel of constant volume. What temperature would be attained theoretically if the gases had an initial temperature of 60° F.?

17. What temperature would calculation lead one to expect when account is taken of the variable specific heat of carbon dioxide?

18. Three pounds of hydrogen are burned in oxygen. (*a*) How much heat is liberated? (*b*) What temperature should be obtained if the specific heats were constant and theoretical oxygen were used, combustion occurring at constant pressure? (*c*) What value should be attained with 25 per cent excess oxygen, variable specific heat and constant pressure?

19. What quantity of heat would be lost by failure to condense the water vapor resulting from the combustion of 3 lbs. of hydrogen if the gases leave at a temperature of 250° F. and room temperature is 60° F.?

20. Determine the approximate higher and lower heat values by Dulong's formulas for a fuel of the following composition by weight: carbon, 70 per cent; hydrogen, 25 per cent; oxygen, 2 per cent; sulphur, 3 per cent.

21. Determine the weight of dry fuel gases per pound of carbon burned for a case in which the gases analyze: carbon dioxide, 14 per cent; carbon monoxide, 2 per cent; hydrogen, 1 per cent; sulphur dioxide, 1 per cent; oxygen, 2 per cent; nitrogen, 80 per cent.

CHAPTER XXX.

1. During the test of a certain boiler it is found that when fired with coal with a heat value of 14,000 B.t.u. per pound the ash contains 0.2 of a pound of carbon per pound of coal fired. (*a*) What is the heat value of the ascending combustible per pound of carbon? (*b*) What is the grate efficiency?

2. If each pound of coal fired as in problem 1 causes the generation of 8 lbs. of dry and saturated steam at a pressure of 150 lbs. abs. from feed

water at a temperature of 120° F., what is the value of the boiler efficiency according to the A.S.M.E. definition?

3. What is the over-all efficiency of the boiler considered in problems 1 and 2 above?

4. If a boiler generates 9 lbs of steam with a quality of 97 per cent at a pressure of 125 lbs. abs. from feed at a temperature of 70° F., what is the equivalent evaporation?

5. A certain boiler generates steam at a pressure of 160 lbs. abs., and superheated 100 degrees from feed at a temperature of 200° F. What is the factor of evaporation?

6. How many pounds of water at a temperature of 70° F. should be converted into dry saturated steam at 125 lbs gauge per hour by a 100-h.p. boiler when operating at normal load? What is the factor of evaporation for this case? How many pounds of material would leave the boiler per hour if it gave steam with a quality of 97 per cent?

CHAPTER XXXII.

1. (*a*) Using Fig. 326 and Table XXIV, how much draft through the boiler will probably be required to burn 20 lbs. of anthracite pea coal per square foot of grate per hour? (*b*) With 50 ft. of flue and two 90° bends, what will be the draft probably required at the base of the stack? (B. and W. Boiler.)

2. (*a*) With flue temperature 550° F. and air at 60°, what would be the theoretical height of stack for case given in problem 1? (*b*) What would be the actual height? (*c*) What would be its diameter in inches for a 2000 boiler h.p. plant?

CHAPTER XXXV.

1. Determine the quantity of heat which will flow per hour between two planes 1 ft. apart and of 6 sq. ins. section with a temperature difference of 100° F., the material being commercial copper and no allowance being made for variation of specific conductivity with temperature. Determine the "heat resistance" of the material between the two planes.

2. Determine the same quantities as called for in problem 1, but for a case in which soft steel is the conducting material.

3. Determine the same quantities as called for in problem 1, but for cases in which water and air are the conducting materials.

4. Tabulate the values obtained for quantity of heat transmitted in above cases and tabulate values as percentage by calling that transmitted by copper 100 per cent.

5. Taking values from Table XXVI, determine the specific conductivity of yellow brass at a temperature of 200° F.

6. Assuming values given in the text as correct, determine the amount of heat lost by radiation in 1 hour from the black surface of a sphere of 1 ft. radius and at a temperature of 1000° F. Do the same for a temperature of 2000° F.

7. (*a*) If the temperature difference at end *a* of the heating surface is 2000° and at end *b* it is 200°, what is the mean temperature difference with flow? (*b*) If $K = 3.7$, how many square feet of heating surface will be required to transmit per hour 33,463 B.t.u. (= 1 boiler h.p.)?

8. A surface condenser receives exhaust steam at temperature 115° F., the initial temperature of the condensing water is 60° F. and its final temperature 105° F. (*a*) What is the mean tenperature difference? (*b*) With $K = 300$, what weight of dry steam will be condensed per square foot of surface per hour? (*c*) What is the efficiency of the surface (neglecting losses)?

9. In a boiler the furnace gases are cooled from 2500° F. to 550° F. and the temperature of the steam is 350° F. (*a*) Neglecting losses, what is the mean temperature difference? (*b*) If $3\frac{1}{2}$ lbs. of steam from and at 212° is evaporated per square foot of heating surface, what is the value of K?

10. With parallel flow the initial and final temperature differences are 500° and 203. (*a*) What is the mean temperature difference? (*b*) With $K = 3$ B.t.u. how much heating surface is required for heating 30 lbs. of water per hour from 60° to 192°?

11. (*a*) With counter-current flow, with a temperature difference of 322° at one end of the heating surface and 357° at the other end, what is the mean temperature difference? (*b*) With $K = 3$ B.t.u. how much heating surface is required for heating 30 lbs. of water per hour from 60° to 192° F.?

12. In a feed water heater 90 lbs. of water per hour are heated by 1 sq. ft. of heating surface, K being 220. (*a*) If the initial temperature difference θ_a is 142°, what is the final value θ_b, neglecting losses? (*b*) What is the efficiency? (*c*) Compute the efficiencies corresponding to different extents of surface and plot curve showing its variations.

13. In a boiler the initial temperature difference between gas and water is 2000° F. (*a*) If one boiler horse-power hour is equivalent to 33,479 B.t.u. and K is 3.7, what will be the final temperature difference (neglecting losses) if 100 lbs. of flue gas (with $C_p = 0.24$) are generated per boiler h.p. hour. (*b*) What is the efficiency? (*c*) Compute the efficiencies corresponding to different extents of surface and plot curves showing its variation.

14. In an economizer with parallel flow, $K = 3$, $W_c = 30$, $W_h = 100$, $C_h = 0.24$, $S = 4$, $T_a = 600$, $t_a = 100$. Find θ_b, T_b and t_b.

15. Same data as problem 14, but for counter flow. (*a*) Find θ_a, θ_b, T_b and t_a.

CHAPTER XXXVI.

1. Determine the theoretical percentage of saving effected by supplying feed water at a temperature of 120° F. instead of 60° F. to a boiler generating dry saturated steam at a pressure of 150 lbs. abs.

2. Determine the theoretical percentage of saving effected by supplying feed water at a temperature of 200° F. instead of 60° F. to a boiler generating dry saturated steam at a pressure of 150 lbs. abs. What is the percentage saving if the boiler superheats the steam 100°? What is the actual amount of heat "saved" per pound of steam generated in each case?

3. How many pounds of steam per pound of water heated will be required to raise the temperature of feed from 60° F to 190° F. in an open heater operated at atmospheric pressure if the steam enters the heater at atmospheric pressure and with 100 per cent quality? How many will be required if the steam has a quality of 90 per cent? What weight of water will leave the heater in each case for every pound of water entering?

4. What is the maximum possible weight of steam exhausted from an engine at atmospheric pressure and 85 per cent quality which could be utilized in an open feed heater if the feed water is to have its temperature raised from 50° F. to 200° F. and if the heater has an efficiency of 95 per cent?

5. An economizer with counter flow receives flue gas at temperature of 600°, and water at 60°. Thirty pounds of water are heated per hour by 100 lbs. of gas ($C_p = 0.24$) through 4 sq. ft. of surface, K being 3. (*a*) Find the increase in the temperature of the feed water and (*b*) the decrease in the temperature of the flue gas.

CHAPTER XXXVII.

1. (*a*) In a direct-contact condenser how much condensing water per pound of steam will be required if the vacuum is 26 ins. and the condensing water is raised from 60° F. to within 10° of the temperature of the exhaust steam ($Ef. = 0.9$)? (*b*) How much is required with 28-in. "vacuum"?

2. (*a*) Same as 1 except for surface condenser, the temperature of the condensate being reduced to 10° below that of the exhaust steam. (*b*) What is the mean temperature difference? (*c*) How much cooling surface is required per pound of steam condensed per hour if $K = 300$?

3. If 400,000 lbs. of condensing water are used per hour in a jet condenser, what would be the probable plunger displacement in cubic feet per minute of a single-acting wet air pump?

4. (*a*) If a surface condenser used with a turbine condenses 10,000 lbs. of steam per hour, what would be the probable plunger displacement of a wet-air pump in cubic feet per minute? (*b*) What, for a dry-air pump, the vacuum being 28 ins. Hg.?

CHAPTER XL.

1 to 6. Same as problems 1–6 under Chap. XXII, but applied to nozzles.

7. (*a*) Neglecting losses, find the discharge velocities per pound of steam flowing in problems 1 to 4. (*b*) Find the area of the nozzle end per pound of steam.

8. With the initial conditions given in problems 1 to 4 plot curves as in Sec. 331.

9. (*a*) If $Ef. = 0.9$, what heat remains per pound of steam at the end of expansion with the same conditions as in problems 1 to 4? (*b*) What is the final quality? (*c*) What is the final entropy?

10. (*a*) Compute the neck areas in problem 8 by Napier's formula. (*b*) Same, by Grashof's formula.

11. (*a*) With uniform flow and allowing 5 lbs. drop in pressure, what would be the diameter of pipe to convey in 1 minute 200 lbs. of dry steam initially at 150 lbs. gauge pressure, the length of pipe being 200 ft.? (*b*) What would be the velocity of flow?

12. (*a*) If a steam engine with cylinder 18 ins. diameter and stroke 24 ins. operates at 200 r.p.m., what should be the diameter of the steam pipe? (*b*) What should be the diameter of the exhaust pipe?

14. (*a*) What weight of air will theoretically flow per second through a nozzle having a neck area of 1 sq. in., if the initial pressure is 80 lbs. per sq. in. abs. and temperature is 60° F. and if the discharge is into the atmosphere? (*b*) What will be the velocity of flow at the neck?

15. (*a*) With initial pressure 20 lbs. per sq. in. abs. and temperature 60° F., what will be the theoretical velocity of discharge to the atmosphere? (*b*) What weight will flow per second through an orifice having an area of 1 sq. in.?

16. With 1 lb. of air flowing through a nozzle per second with initial pressure (P_1) 80 lbs. per sq. in. abs. and temperature 60° F., plot curves showing how the decrease in back pressure (P_x) affects (*a*) the velocity of flow, (*b*) the volume of the air and (*c*) the area of the nozzle, the abscissas being ratios P_1/P_x.

CHAPTER XLI.

1. Determine the work which must be done per cycle in an air compressor cylinder without clearance which operates under the following ideal conditions. At the end of the suction stroke the cylinder contains one-tenth of a pound of air at a temperature of 60° F. and a pressure of 14.7 lbs. per sq. in. abs.

2. Determine the work which must be done per cycle in an air compressor cylinder with clearance equal to 5 per cent of the piston displacement and which operates under the following conditions. It draws in one-tenth of a pound of air during the suction stroke; this air mixed with that caught in the clearance has a temperature of 60° F. at the end of the suction stroke; the pressure during the suction stroke is constant and equal to 14.7 lbs. per sq. in. abs.; compression and expansion are adiabatic; and air is discharged at a constant pressure of 50 lbs. per sq. in. abs.

3. Determine the capacity of such a compressor in terms of free air (60° F. and 14.7 lbs.) per min. if it operates at a speed of 180 r.p.m. and is built double acting.

4. Assume three compressor cylinders without clearance and with a piston displacement of 1 cu. ft., one cylinder so arranged as to give adiabatic compression; one arranged to give isothermal compression; and one arranged to

give a compression curve with exponent equal to 1.25. (*a*) Determine the work done during one compression in each cylinder and the final temperature in each case if air with an initial temperature of 60° F. and at an initial pressure of 14.7 lbs. per sq. in. abs. is compressed to 45 lbs. per sq. in. abs. (*b*) Express the saving in the second and third cases as a percentage of the compression work in the case of the adiabatic process. (*c*) Determine the work per cycle in each case, assuming discharge to occur at the constant pressure of 45 lbs. (*d*) Express savings as per cent, as in (*b*). (*e*) Make calculations called for in (*a*), (*b*), (*c*) and (*d*) but with a discharge pressure of 100 lbs. per sq. in. abs.

5. Compare the work done in the air cylinder per cycle in the following cases, express saving as per cent of work in least favorable case, and determine discharge temperature. (*a*) An air compressor cylinder without clearance has a diameter of 16 ins. and a stroke of 18 ins. The air at the end of the suction stroke has a temperature of 60° F., and a pressure of 14.7 lbs. per sq. in. abs. The compression is adiabatic and discharge occurs at a constant pressure of 100 lbs. per sq. in. abs. (*b*) The same cylinder with the same initial conditions gives a compression line with exponent equal to 1.3. (*c*) Two-stage compression is used with an intercooler. The low-pressure cylinder has the same size as before and operates under the same initial conditions but gives a compression curve with exponent equal to 1.22 and discharges at such a pressure that cooling at constant pressure to initial temperature will give the air a volume equal to the piston displacement of the high-pressure cylinder. The high-pressure cylinder has a diameter of 10 ins. and a stroke of 18 ins., gives a compression curve with exponent equal to 1.22 and discharges at a constant pressure of 100 lbs. per sq. in. There are no transfer losses between stages.

6. Assume that 1 lb of air with initial conditions 60° F. and 14.7 lbs. per sq. in. abs. has been compressed adiabatically to a pressure of 80 lbs. per sq. in., and then cooled at constant pressure to initial temperature after discharge from the compressor. If this air is used in an air engine without clearance, operating on a cycle of the same shape as the PV-diagram of the Clausius cycle, and expanding to atmospheric pressure, determine (*a*) the work made available in the engine. (*b*) the efficiency of the process in the ideal case, *i.e.*, work made available ÷ work done in air compressor cylinder. (*c*) The final temperature attained in the engine cylinder.

7. Assume the same conditions as in problem 6 above, excepting that the air before entrance to the engine is preheated at constant pressure to a temperature of 300° F. Determine the values called for under (*a*), (*b*) and (*c*) of that problem.

APPENDIX.

USE OF COMMON LOGARITHMS FOR SPECIAL CASES.

Case I. To Determine the nth Power of a Number Less than Unity.

Example: Find $0.5^{1.55}$ by logs.

In general $\log_{10} V^n = n \log_{10} V$; and in this case $V = 0.5$ and $n = 1.55$.

From the tables $\log_{10} 0.5 = 9.6990 - 10$,

Then, $1.55 \log_{10} 0.5 = 1.55\ (9.6990 - 10) = 15.033450 - 15.5$

Subtract 5.5 to reduce the negative part of the characteristic to 10, $\underline{5.5 \qquad - \ 5.5}$

Log. of answer = $9.533450 - 10.0$

Corresponding number = $0.342 = 0.5^{1.55}$.

(Note that a fraction raised to a power greater than unity gives a result less than the original fraction.)

Case II. To Determine the nth Root of a Fraction.

Example: Given $V^{1.5} = 0.75$; Find V. Evidently, —

$$\log_{10} V = \log_{10} \left(\sqrt[1.5]{0.75}\right) = \log_{10} \left(0.75^{\frac{1}{1.5}}\right) = (\log_{10} 0.75) \div 1.5,$$

which is in the general form of $\log_{10} V = (\log_{10} C) \div n$,

where $C = 0.75$ and $n = 1.5$.

From the tables $\log_{10} 0.75 = 9.8751 - 10$.

Then $$\frac{(\log 0.75)}{1.5} = \frac{(9.8751 - 10)}{1.5} = 6.5833 - 6.666.$$

Add 3.334 to raise the negative part of the characteristic to 10, $\underline{3.3340 - \ 3.334}$

Log. of $V = 9.9173 - 10.000$

The corresponding number is 0.8266 which is $\sqrt[1.5]{0.75}$.

TABLE A. — COMMON LOGARITHMS (Log_{10}).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
0	0	0000	3010	4771	6021	6990	7782	8451	9031	9542	
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	42
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	38
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	35
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	32
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	30
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	28
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	26
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	25
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	24
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	22
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	21
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	20
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	19
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	19
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	18
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	17
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	16
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	15
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	14
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	13
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	12
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	11
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	11
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	10
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	10
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	9
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	9
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	9
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	9
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	8
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	8
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	8

$e = 2.71828$

TABLE A. (*Concluded*). — COMMON LOGARITHMS (Log_{10}).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	8
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	8
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	8
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	7
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	7
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	7
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	7
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	7
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	7
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	7
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	6
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	6
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	6
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	6
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	6
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	6
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	6
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	6
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	5
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	5
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	5
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	5
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	5
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	5
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	5
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	5
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	5
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	4

Naperian $\log_e = 2.302 \log_{10}$.

TABLE B. — HYPERBOLIC OR NAPERIAN LOGARITHMS, ($\log_e$).

N.	Log.	N.	Log.	N.	Log.	N.	Log.	N.	Log.
1.00	0.0000	2.30	0.8329	3.60	1.2809	4.90	1.5892	6.40	1.8563
1.05	0.0488	2.35	0.8544	3.65	1.2947	4.95	1.5994	6.50	1.8718
1.10	0.0953	2.40	0.8755	3.70	1.3083	5.00	1.6094	6.60	1.8871
1.15	0.1398	2.45	0.8961	3.75	1.3218	5.05	1.6194	6.70	1.9021
1.20	0.1823	2.50	0.9163	3.80	1.3350	5.10	1.6292	6.80	1.9169
1.25	0.2231	2.55	0.9361	3.85	1.3481	5.15	1.6390	6.90	1.9315
1.30	0.2624	2.60	0.9555	3.90	1.3610	5.20	1.6487	7.00	1.9459
1.35	0.3001	2.65	0.9746	3.95	1.3737	5.25	1.6582	7.20	1.9741
1.40	0.3365	2.70	0.9933	4.00	1.3863	5.30	1.6677	7.40	2.0015
1.45	0.3716	2.75	1.0116	4.05	1.3987	5.35	1.6771	7.60	2.0281
1.50	0.4055	2.80	1.0296	4.10	1.4110	5.40	1.6864	7.80	2.0541
1.55	0.4383	2.85	1.0473	4.15	1.4231	5.45	1.6956	8.00	2.0794
1.60	0.4700	2.90	1.0647	4.20	1.4351	5.50	1.7047	8.25	2.1102
1.65	0.5008	2.95	1.0818	4.25	1.4469	5.55	1.7138	8.50	2.1401
1.70	0.5306	3.00	1.0986	4.30	1.4586	5.60	1.7228	8.75	2.1691
1.75	0.5596	3.05	1.1151	4.35	1.4702	5.65	1.7317	9.00	2.1972
1.80	0.5878	3.10	1.1314	4.40	1.4816	5.70	1.7405	9.25	2.2246
1.85	0.6152	3.15	1.1474	4.45	1.4929	5.75	1.7492	9.50	2.2513
1.90	0.6419	3.20	1.1632	4.50	1.5041	5.80	1.7579	9.75	2.2773
1.95	0.6678	3.25	1.1787	4.55	1.5151	5.85	1.7664	10.00	2.3026
2.00	0.6931	3.30	1.1939	4.60	1.5261	5.90	1.7750	11.00	2.3979
2.05	0.7178	3.35	1.2090	4.65	1.5369	5.95	1.7834	12.00	2.4849
2.10	0.7419	3.40	1.2238	4.70	1.5476	6.00	1.7918	13.00	2.5649
2.15	0.7655	3.45	1.2384	4.75	1.5581	6.10	1.8083	14.00	2.6391
2.20	0.7885	3.50	1.2528	4.80	1.5686	6.20	1.8245	15.00	2.7081
2.25	0.8109	3.55	1.2669	4.85	1.5790	6.30	1.8405	16.00	2.7726

(1) To find $\log_e$ of a number greater than 10 (for example 21): —
$\text{Log}_e 21 = \log_e (10 \times 2.1) = \log_e 10 + \log_e 2.1 = 2.3026 + 0.7419 = 3.0445$.
(2) Base $e = 2.71828$ and $\log_e a = 2.302 \times \log_{10} a$.

TABLE C.

Giving the Properties of One Pound of

SATURATED STEAM.

[Condensed and rearranged from Marks and Davis's Steam Tables and Diagrams, 1909, by permission of the publishers, Longmans, Green & Co., with addition of lines and columns as indicated by asterisks (*). The first eleven lines constitute a table based on temperatures which are so selected as to give (approximately) intervals of half an inch of Hg, in column 1, without using fractional degrees. The rest of the table is based on absolute pressures. At the heads of columns are given the alternative symbols which are often used.]

Pressure.		Temp. °F.	Heat — British Thermal Units.					Entropy.			Sp. Vol. Cu. Ft.	Pres. Abs. Lbs. Sq. In.
Inches of Mercury.	Abs. Lbs. Sq. In.		Total.	Water.	Latent.			Total.	Liquid.	Vapor.		
					Total.	Internal.	External.					
	p	t	$\lambda\,(H)$	$q\,(h)$	$r\,(L)$	$\rho\,(I)$	APu*	$\Delta\phi\,(N)$	$\Delta\phi_l\,(n)\,(\theta)$	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	$V\,(v)\,(s)$	p
0.1804	0.0886	32	1073.4	0.0	1073.4	1019.3	54.1	2.1832	0.0000	2.1832	3294.0	
0.50+	0.2472	59	1085.4	27.08	1058.3	1001.3	57.0	2.0944	0.0536	2.0408	1249.2	
1.00−	0.4893	79	1094.3	47.04	1047.3	988.1	59.2	2.0358	0.0913	1.9445	657.0	
1.50+	0.741	92	1100.1	60.0	1040.0	979.4	60.6	2.0007	0.1151	1.8856	442.2	
2.00−	0.975	101	1104.0	68.97	1035.1	973.5	61.6	1.9776	0.1313	1.8463	341.0	
2.50+	1.235	109	1107.5	76.94	1030.6	968.2	62.4	1.9678	0.1454	1.8124	272.9	
3.00−	1.467	115	1110.2	82.92	1027.2	964.2	63.0	1.9435	0.1559	1.7876	231.9	
3.50+	1.736	121	1112.8	88.91	1023.9	960.2	63.7	1.9296	0.1662	1.7634	197.9	
4.00+	1.992	126	1115.0	93.90	1021.1	956.8	64.3	1.9183	0.1747	1.7436	173.9	
4.50+	2.219	130	1116.7	97.89	1018.8	954.1	64.7	1.9095	0.1816	1.7279	157.1	
5.00+	2.467	134	1118.4	101.88	1016.5	951.4	65.1	1.9008	0.1883	1.7125	142.2	
	1	101.83	1104.4	69.8	1034.6	972.9	61.7	1.9754	0.1327	1.8427	333.0	1
	2	126.15	1115.0	94.0	1021.0	956.7	64.3	1.9180	0.1749	1.7431	173.5	2
	3	141.52	1121.6	109.4	1012.3	946.4	65.9	1.8848	0.2008	1.6840	118.5	3
	4	153.01	1126.5	120.9	1005.7	938.6	67.1	1.8614	0.2198	1.6416	90.5	4
	5	162.28	1130.5	130.1	1000.3	932.4	67.9	1.8432	0.2348	1.6084	73.33	5

TABLE C.—SATURATED STEAM (*Continued*).

Pressure.		Temp. °F.	Heat — British Thermal Units.					Entropy.			Sp. Vol. Cu. Ft.	Pres. Abs. Lbs. Sq. In.
* Gauge Pressure. Lbs. Sq. In.	Abs. Lbs. Sq. In.		Total.	Water.	Latent.			Total.	Liquid.	Vapor.		
					Total.	Internal.	External.					
	p	t	λ (H)	q (h)	r (L)	ρ (I)	APu*	$\Delta\phi$ (N)	$\Delta\phi_l$ (n) (θ)	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	V (v) (s)	p
	6	170.06	1133.7	137.9	995.8	927.0	68.8	1.8285	0.2471	1.5814	61.89	6
	7	176.85	1136.5	144.7	991.8	922.4	69.4	1.8161	0.2579	1.5582	53.56	7
	8	182.86	1139.0	150.8	988.2	918.2	70.0	1.8053	0.2673	1.5380	47.27	8
	9	188.27	1141.1	156.2	985.0	914.4	70.6	1.7958	0.2756	1.5202	42.36	9
	10	193.22	1143.1	161.1	982.0	910.9	71.1	1.7874	0.2832	1.5042	38.38	10
	11	197.75	1144.9	165.7	979.2	907.8	71.4	1.7797	0.2902	1.4895	35.10	11
	12	201.96	1146.5	169.9	976.6	904.8	71.8	1.7727	0.2967	1.4760	32.36	12
	13	205.87	1148.0	173.8	974.2	902.0	72.2	1.7664	0.3025	1.4639	30.03	13
	14	209.55	1149.4	177.5	971.9	899.3	72.6	1.7604	0.3081	1.4523	28.02	14
*0	14.7	212.00	1150.4	180.00	970.4	897.6	72.8	1.7565	0.3118	1.4447	26.79	14.7
0.3	15	213.0	1150.7	181.0	969.7	896.8	72.9	1.7549	0.3133	1.4416	26.27	15
1.3	16	216.3	1152.0	184.4	967.6	894.4	73.2	1.7494	0.3183	1.4311	24.79	16
2.3	17	219.4	1153.1	187.5	965.6	892.1	73.5	1.7444	0.3229	1.4215	23.38	17
3.3	18	222.4	1154.2	190.5	963.7	889.9	73.8	1.7400	0.3273	1.4127	22.16	18
4.3	19	225.2	1155.2	193.4	961.8	887.8	74.0	1.7360	0.3315	1.4045	21.07	19
5.3	20	228.0	1156.2	196.1	960.0	885.8	74.2	1.7320	0.3355	1.3965	20.08	20
6.3	21	230.6	1157.1	198.8	958.3	883.9	74.4	1.7280	0.3393	1.3887	19.18	21
7.3	22	233.1	1158.0	201.3	956.7	882.0	74.7	1.7241	0.3430	1.3811	18.37	22
8.3	23	235.5	1158.8	203.8	955.1	880.2	74.9	1.7204	0.3465	1.3739	17.62	23
9.3	24	237.8	1159.6	206.1	953.5	878.5	75.0	1.7169	0.3499	1.3670	16.93	24
10.3	25	240.1	1160.4	208.4	952.0	876.8	75.2	1.7136	0.3532	1.3604	16.30	25

TABLE C.—SATURATED STEAM (*Continued*).

Pressure. * Gauge Pressure. Lbs. Sq. In.	Pressure. Abs. Lbs. Sq. In.	Temp. °F.	Heat — British Thermal Units. Total.	Water.	Latent. Total.	Latent. Internal.	Latent. External.	Entropy. Total.	Entropy. Liquid.	Entropy. Vapor.	Sp. Vol. Cu. Ft.	Pres. Abs. Lbs. Sq. In.
	p	t	λ (H)	q (h)	r (L)	ρ (I)	APu*	$\Delta\phi$ (N)	$\Delta\phi_l$ (n) (θ)	$\Delta\phi_r\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	V (v) (s)	p
11.3	26	242.2	1161.2	210.6	950.6	875.1	75.5	1.7106	0.3564	1.3542	15.72	26
12.3	27	244.4	1161.9	212.7	949.2	873.5	75.7	1.7077	0.3594	1.3483	15.18	27
13.3	28	246.4	1162.6	214.8	947.8	872.0	75.8	1.7048	0.3623	1.3425	14.67	28
14.3	29	248.4	1163.2	216.8	946.4	870.5	75.9	1.7019	0.3652	1.3367	14.19	29
15.3	30	250.3	1163.9	218.8	945.1	869.0	76.1	1.6991	0.3680	1.3311	13.74	30
17.3	32	254.1	1165.1	222.6	942.5	866.2	76.3	1.6938	0.3733	1.3205	12.93	32
19.3	34	257.6	1166.3	226.2	940.1	863.4	76.7	1.6891	0.3784	1.3107	12.22	34
21.3	36	261.0	1167.3	229.6	937.7	860.8	76.9	1.6846	0.3832	1.3014	11.58	36
23.3	38	264.2	1168.4	232.9	935.5	858.3	77.2	1.6802	0.3877	1.2925	11.01	38
25.3	40	267.3	1169.4	236.1	933.3	855.9	77.4	1.6761	0.3920	1.2841	10.49	40
27.3	42	270.2	1170.3	239.1	931.2	853.6	77.6	1.6721	0.3962	1.2759	10.02	42
29.3	44	273.1	1171.2	242.0	929.2	851.3	77.9	1.6683	0.4002	1.2681	9.59	44
31.3	46	275.8	1172.0	244.8	927.2	849.2	78.0	1.6647	0.4040	1.2607	9.20	46
33.3	48	278.5	1172.8	247.5	925.3	847.1	78.2	1.6613	0.4077	1.2536	8.84	48
35.3	50	281.0	1173.6	250.1	923.5	845.0	78.5	1.6581	0.4113	1.2468	8.51	50
37.3	52	283.5	1174.3	252.6	921.7	843.1	78.6	1.6549	0.4147	1.2402	8.20	52
39.3	54	285.9	1175.0	255.1	919.9	841.1	78.8	1.6519	0.4180	1.2339	7.91	54
41.3	56	288.2	1175.7	257.5	918.2	839.3	78.9	1.6490	0.4212	1.2278	7.65	56
43.3	58	290.5	1176.4	259.8	916.5	837.4	79.1	1.6460	0.4242	1.2218	7.40	58
45.3	60	292.7	1177.0	262.1	914.9	835.6	79.3	1.6432	0.4272	1.2160	7.17	60

TABLE C — SATURATED STEAM (*Continued*).

PRESSURE.		TEMP. °F.	HEAT — British Thermal Units.					ENTROPY.			SP. VOL. Cu. Ft.	PRES. ABS. Lbs. Sq. In.
			Total.	Water.	Latent.			Total.	Liquid.	Vapor.		
					Total.	Internal.	External.					
* Gauge Pressure. Lbs. Sq. In.	ABS. Lbs. Sq. In.											
	p	t	λ (H)	q (h)	r (L)	ρ (I)	APu*	$\Delta\phi$ (N)	$\Delta\phi_l$ (n) (θ)	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	V (v) (s)	p
47.3	62	294.9	1177.6	264.3	913.3	833.9	79.4	1.6406	0.4302	1.2104	6.95	62
49.3	64	297.0	1178.2	266.4	911.8	832.2	79.6	1.6380	0.4330	1.2050	6.75	64
51.3	66	299.0	1178.8	268.5	910.2	830.5	79.7	1.6355	0.4358	1.1998	6.56	66
53.3	68	301.0	1179.3	270.6	908.7	828.9	79.8	1.6331	0.4385	1.1946	6.38	68
55.3	70	302.9	1179.8	272.6	907.2	827.3	79.9	1.6307	0.4411	1.1896	6.20	70
57.3	72	304.8	1180.4	274.5	905.8	825.8	80.0	1.6285	0.4437	1.1848	6.04	72
59.3	74	306.7	1180.9	276.5	904.4	824.2	80.2	1.6263	0.4462	1.1801	5.89	74
61.3	76	308.5	1181.4	278.3	903.0	822.7	80.3	1.6242	0.4487	1.1755	5.74	76
63.3	78	310.3	1181.8	280.2	901.7	821.3	80.4	1.6221	0.4511	1.1710	5.60	78
65.3	80	312.0	1182.3	282.0	900.3	819.8	80.5	1.6200	0.4535	1.1665	5.47	80
67.3	82	313.8	1182.8	283.8	899.0	818.4	80.6	1.6180	0.4557	1.1623	5.34	82
69.3	84	315.4	1183.2	285.5	897.7	817.0	80.7	1.6160	0.4579	1.1581	5.22	84
71.3	86	317.1	1183.6	287.2	896.4	815.6	80.8	1.6141	0.4601	1.1540	5.10	86
73.3	88	318.7	1184.0	288.9	895.2	814.3	80.9	1.6123	0.4623	1.1500	5.00	88
75.3	90	320.3	1184.4	290.5	893.9	813.0	80.9	1.6105	0.4644	1.1461	4.89	90
77.3	92	321.8	1184.8	292.1	892.7	811.7	81.0	1.6087	0.4664	1.1423	4.79	92
79.3	94	323.4	1185.2	293.7	891.5	810.4	81.1	1.6069	0.4684	1.1385	4.69	94
81.3	96	324.9	1185.6	295.3	890.3	809.1	81.2	1.6052	0.4704	1.1348	4.60	96
83.3	98	326.4	1186.0	296.8	889.2	807.9	81.3	1.6036	0.4724	1.1312	4.51	98
85.3	100	327.8	1186.3	298.3	888.0	806.6	81.4	1.6020	0.4743	1.1277	4.429	100

TABLE C. — SATURATED STEAM (*Continued*).

PRESSURE.		TEMP. °F.	HEAT — British Thermal Units.					ENTROPY.			SP. VOL. Cu. Ft.	PRES. ABS. Lbs. Sq. In.
* Gauge Pressure Lbs. Sq. In.	ABS. Lbs. Sq. In.		Total.	Water.	Latent.			Total.	Liquid.	Vapor.		
					Total.	Internal.	External.					
	p	t	$\lambda\,(H)$	q (h)	r (L)	$\rho\,(I)$	APu*	$\Delta\phi\,(N)$	$\Delta\phi_l\,(n)\,(\theta)$	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	V $(v)\,(s)$	p
90.3	105	331.4	1187.2	302.0	885.2	803.6	81.6	1.5980	0.4789	1.1191	4.230	105
95.3	110	334.8	1188.0	305.5	882.5	800.7	81.8	1.5942	0.4834	1.1108	4.047	110
100.3	115	338.1	1188.8	309.0	879.8	797.9	81.9	1.5907	0.4877	1.1030	3.880	115
105.3	120	341.3	1189.6	312.3	877.2	795.2	82.0	1.5873	0.4919	1.0954	3.726	120
110.3	125	344.4	1190.3	315.5	874.7	792.6	82.1	1.5839	0.4959	1.0880	3.583	125
115.3	130	347.4	1191.0	318.6	872.3	790.0	82.3	1.5807	0.4998	1.0809	3.452	130
120.3	135	350.3	1191.6	321.7	869.9	787.5	82.4	1.5777	0.5035	1.0742	3.331	135
125.3	140	353.1	1192.2	324.6	867.6	785.0	82.6	1.5747	0.5072	1.0675	3.219	140
130.3	145	355.8	1192.8	327.4	865.4	782.7	82.7	1.5719	0.5107	1.0612	3.112	145
135.3	150	358.5	1193.4	330.2	863.2	780.4	82.8	1.5692	0.5142	1.0550	3.012	150
140.3	155	361.0	1194.0	332.9	861.0	778.1	82.9	1.5664	0.5175	1.0489	2.920	155
145.3	160	363.6	1194.5	335.6	858.8	775.8	83.0	1.5639	0.5208	1.0431	2.834	160
150.3	165	366.0	1195.0	338.2	856.8	773.6	83.2	1.5615	0.5239	1.0376	2.753	165
155.3	170	368.5	1195.4	340.7	854.7	771.5	83.2	1.5590	0.5269	1.0321	2.675	170
160.3	175	370.8	1195.9	343.2	852.7	769.4	83.3	1.5567	0.5299	1.0268	2.602	175
165.3	180	373.1	1196.4	345.6	850.8	767.4	83.4	1.5543	0.5328	1.0215	2.533	180
170.3	185	375.4	1196.8	348.0	848.8	765.4	83.4	1.5520	0.5356	1.0164	2.468	185
175.3	190	377.6	1197.3	350.4	846.9	763.4	83.5	1.5498	0.5384	1.0114	2.406	190
180.3	195	379.8	1197.7	352.7	845.0	761.4	83.6	1.5476	0.5410	1.0066	2.346	195
185.3	200	381.9	1198.1	354.9	843.2	759.5	83.7	1.5456	0.5437	1.0019	2.290	200

TABLE C. — SATURATED STEAM (*Concluded*).

PRESSURE.		TEMP. °F.	HEAT — British Thermal Units.					ENTROPY.			SP. VOL. Cu. Ft.	PRES. ABS. Lbs. Sq. In.
			Total.	Water.	Latent.			Total.	Liquid.	Vapor.		
*Gauge Pressure Lbs. Sq. In.	ABS. Lbs. Sq. In.				Total.	Internal.	External.					
	p	t	$\lambda\,(H)$	q (h)	r (L)	$\rho\,(I)$	APu*	$\Delta\phi\,(N)$	$\Delta\phi_l\,(n)\,(\theta)$	$\Delta\phi_v\left(\frac{r}{T}\right)\left(\frac{L}{T}\right)$	V $(v)\,(s)$	p
235.3	250	401.1	1201.5	375.2	826.3	742.0	84.3	1.5276	0.5676	0.9600	1.850	250
285.3	300	417.5	1204.1	392.7	811.3	726.8	84.5	1.5129	0.5878	0.9251	1.551	300
385.3	400	444.8	1208	422	786.0	701	85	1.489	0.621	0.868	1.17	400
485.3	500	467.3	1210	448	762	678	84	1.470	0.648	0.822	0.93	500
*985.3	1000	545	1200	535	665	587	78	1.39	0.73	0.66	0.4	1000
*	2000	640	?	?	490	430	60	?	?	0.45	0.2	2000
* Crit.	3200	706	?	?	0	0	0	?	?	?	0.05	3200

TABLE D

PROPERTIES OF ONE POUND OF

SUPERHEATED STEAM.

[Condensed from Marks and Davis's STEAM TABLES AND DIAGRAMS, 1909, by permission of the publishers, Longmans, Green & Co.]

$Sp.V$ = specific volume in cu. ft.; ΔQ = B.t.u. total heat above 32° F.; $\Delta\phi$ = total entropy above 32° F.

Absolute Pressure. Lbs. Sq. In. / Sat. Temp. °F.		Degrees of Superheat. 0	50	100	150	200	250	300
15 (213)	Sp. V.	26.27	28.40	30.46	32.50	34.53	36.56	38.58
	ΔQ	1150.7	1174.2	1197.6	1221.0	1244.4	1267.7	1291.1
	$\Delta\phi$	1.7549	1.7886	1.8199	1.8492	1.8768	1.9029	1.9276
50 (281)	Sp. V.	8.51	9.19	9.84	10.48	11.11	11.74	12.36
	ΔQ	1173.6	1198.8	1223.4	1247.7	1271.8	1295.8	1319.7
	$\Delta\phi$	1.6581	1.6909	1.7211	1.7491	1.7755	1.8002	1.8237
100 (327.8)	Sp. V.	4.43	4.79	5.14	5.47	5.80	6.12	6.44
	ΔQ	1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8
	$\Delta\phi$	1.6020	1.6358	1.6658	1.6933	1.7188	1.7428	1.7656
110 (334.8)	Sp. V.	4.05	4.38	4.70	5.01	5.31	5.61	5.90
	ΔQ	1188.0	1215.9	1242.0	1267.1	1291.9	1316.2	1340.4
	$\Delta\phi$	1.5942	1.6282	1.6583	1.6857	1.7110	1.7350	1.7576
120 (341.3)	Sp. V.	3.73	4.04	4.33	4.62	4.89	5.17	5.44
	ΔQ	1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7
	$\Delta\phi$	1.5873	1.6216	1.6517	1.6789	1.7041	1.7280	1.7505
130 (347.4)	Sp. V.	3.45	3.74	4.02	4.28	4.54	4.80	5.05
	ΔQ	1191.0	1219.7	1246.1	1271.4	1296.2	1320.6	1344.9
	$\Delta\phi$	1.5807	1.6153	1.6453	1.6724	1.6976	1.7213	1.7437
140 (353.1)	Sp. V.	3.22	3.49	3.75	4.00	4.24	4.48	4.71
	ΔQ	1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9
	$\Delta\phi$	1.5747	1.6096	1.6395	1.6666	1.6916	1.7152	1.7376
150 (358.5)	Sp. V.	3.01	3.27	3.51	3.75	3.97	4.19	4.41
	ΔQ	1193.4	1223.0	1249.6	1275.1	1300.0	1324.5	1348.8
	$\Delta\phi$	1.5692	1.6043	1.6343	1.6612	1.6862	1.7097	1.7320
160 (363.6)	Sp. V.	2.83	3.07	3.30	3.53	3.74	3.95	4.15
	ΔQ	1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6
	$\Delta\phi$	1.5693	1.5993	1.6292	1.6561	1.6810	1.7043	1.7266
170 (368.5)	Sp. V.	2.68	2.91	3.12	3.34	3.54	3.73	3.92
	ΔQ	1195.4	1225.9	1252.8	1278.4	1303.3	1327.9	1352.3
	$\Delta\phi$	1.5590	1.5947	1.6246	1.6513	1.6762	1.6994	1.7217
180 (373.1)	Sp. V.	2.53	2.75	2.96	3.16	3.35	3.54	3.72
	ΔQ	1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9
	$\Delta\phi$	1.5543	1.5904	1.6201	1.6468	1.6716	1.6948	1.7169
190 (377.6)	Sp. V.	2.41	2.62	2.81	3.00	3.19	3.37	3.55
	ΔQ	1197.3	1228.6	1255.7	1281.3	1306.3	1330.9	1355.5
	$\Delta\phi$	1.5498	1.5862	1.6159	1.6425	1.6627	1.6904	1.7124
200 (381.9)	Sp. V.	2.29	2.49	2.68	2.86	3.04	3.21	3.38
	ΔQ	1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0
	$\Delta\phi$	1.5456	1.5823	1.6120	1.6385	1.6632	1.6862	1.7082
300 (417.5)	Sp. V.	1.55	1.69	1.83	1.96	2.09	2.21	2.33
	ΔQ	1204.1	1240.3	1268.2	1294.0	1319.3	1344.3	1369.2
	$\Delta\phi$	1.5129	1.5530	1.5824	1.6082	1.6323	1.6550	1.6765
500 (467.3)	Sp. V.	0.93	1.03	1.113	1.22	1.31	1.39	1.47
	ΔQ	1210.0	1256	1285	1311	1337	1362	1388
	$\Delta\phi$	1.470	1.519	1.548	1.573	1.597	1.619	1.640

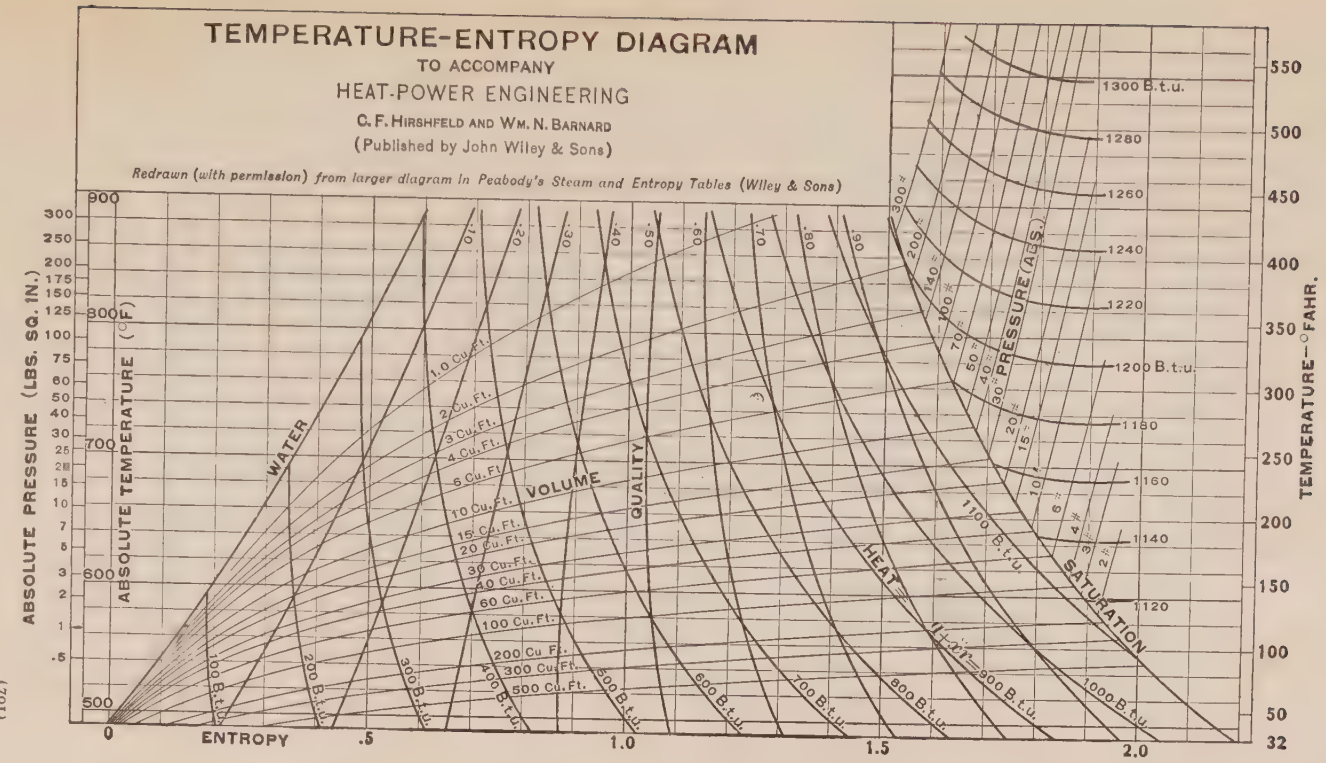

TEMPERATURE-ENTROPY DIAGRAM
TO ACCOMPANY
HEAT-POWER ENGINEERING
C. F. HIRSHFELD AND WM. N. BARNARD
(Published by John Wiley & Sons)
Redrawn (with permission) from larger diagram in Peabody's Steam and Entropy Tables (Wiley & Sons)
ABSOLUTE PRESSURE (LBS. SQ. IN.)
ABSOLUTE TEMPERATURE (°F.)
TEMPERATURE—°FAHR.
ENTROPY
WATER
VOLUME
QUALITY
HEAT
SATURATION
PRESSURE (ABS.)
1.0 Cu. Ft.
2 Cu. Ft.
3 Cu. Ft.
4 Cu. Ft.
6 Cu. Ft.
10 Cu. Ft.
15 Cu. Ft.
20 Cu. Ft.
30 Cu. Ft.
40 Cu. Ft.
60 Cu. Ft.
100 Cu. Ft.
200 Cu. Ft.
300 Cu. Ft.
500 Cu. Ft.
100 B.t.u.
200 B.t.u.
300 B.t.u.
400 B.t.u.
500 B.t.u.
600 B.t.u.
700 B.t.u.
800 B.t.u.
900 B.t.u.
1000 B.t.u.
1100 B.t.u.
1200 B.t.u.
1300 B.t.u.
1280
1260
1240
1220
1180
1160
1140
1120

PLATE I.

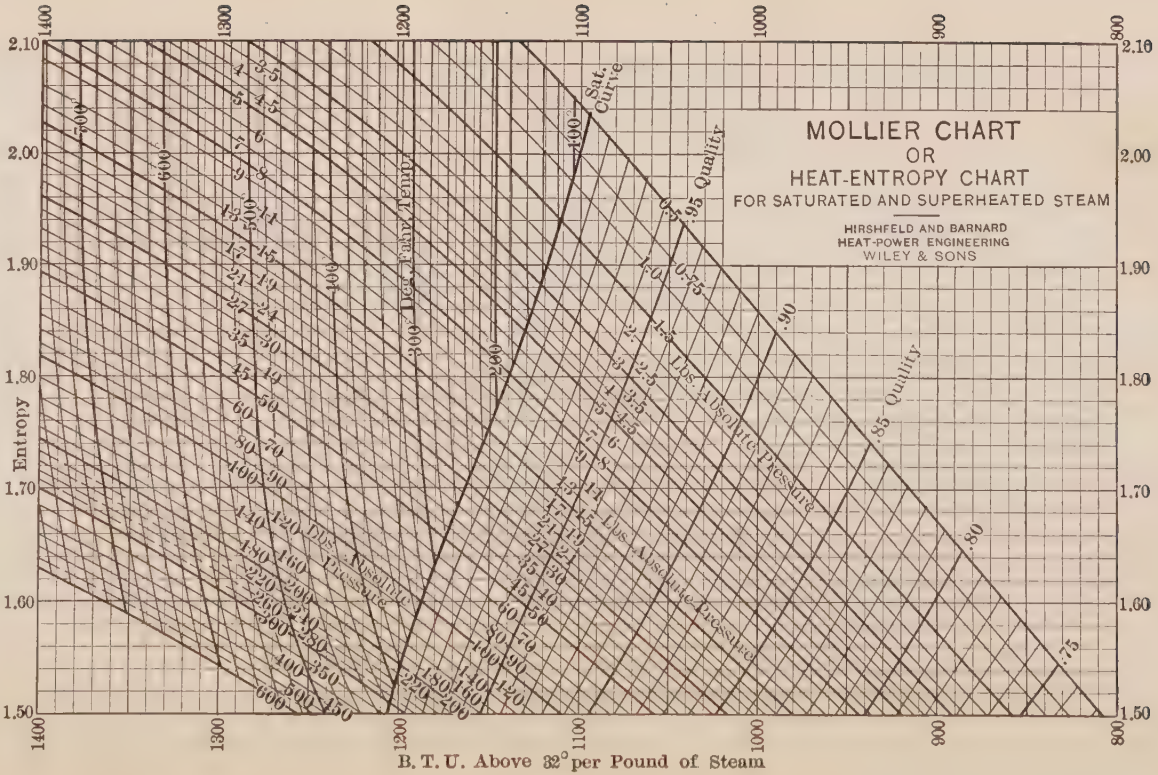

MOLLIER CHART
OR
HEAT-ENTROPY CHART
FOR SATURATED AND SUPERHEATED STEAM
HIRSHFELD AND BARNARD
HEAT-POWER ENGINEERING
WILEY & SONS
Entropy
B. T. U. Above 32° per Pound of Steam
Sat. Curve
Quality
Lbs. Absolute Pressure
Lbs. Absolute Pressure
Lbs. Absolute Pressure

PLATE II.

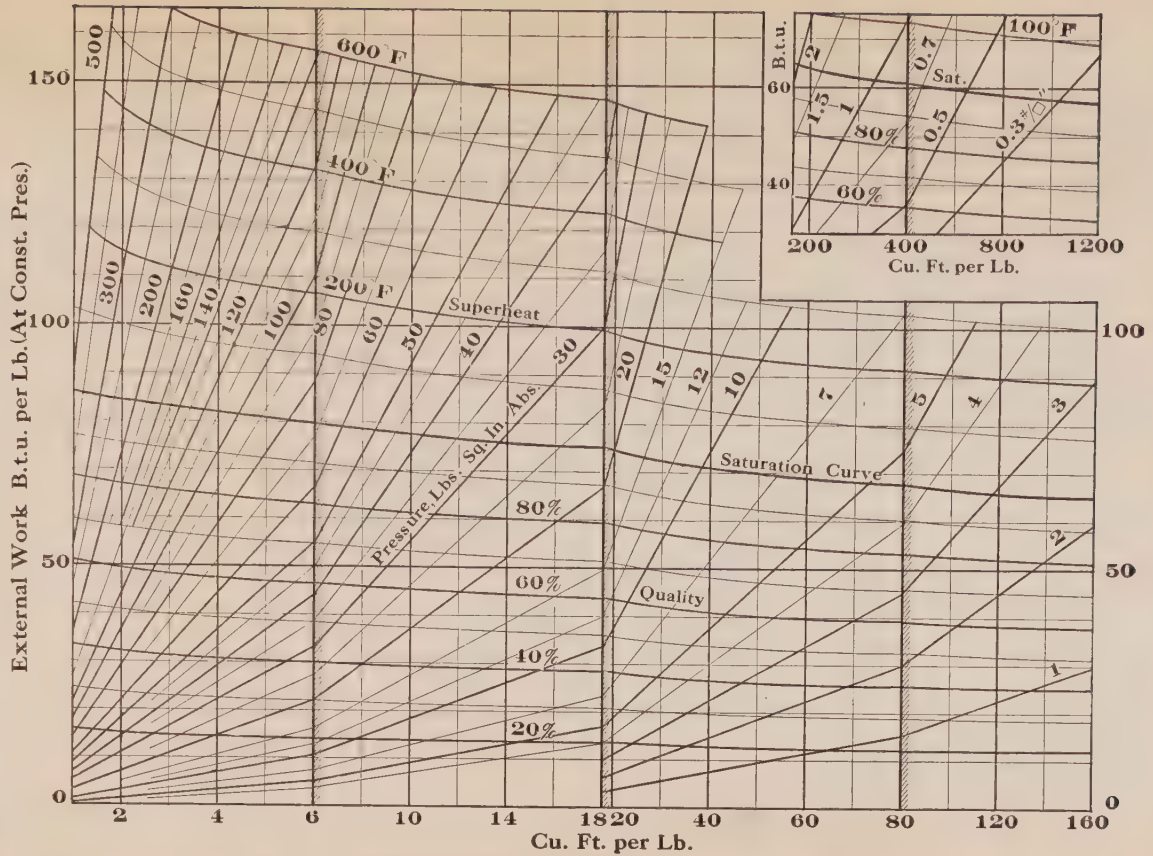
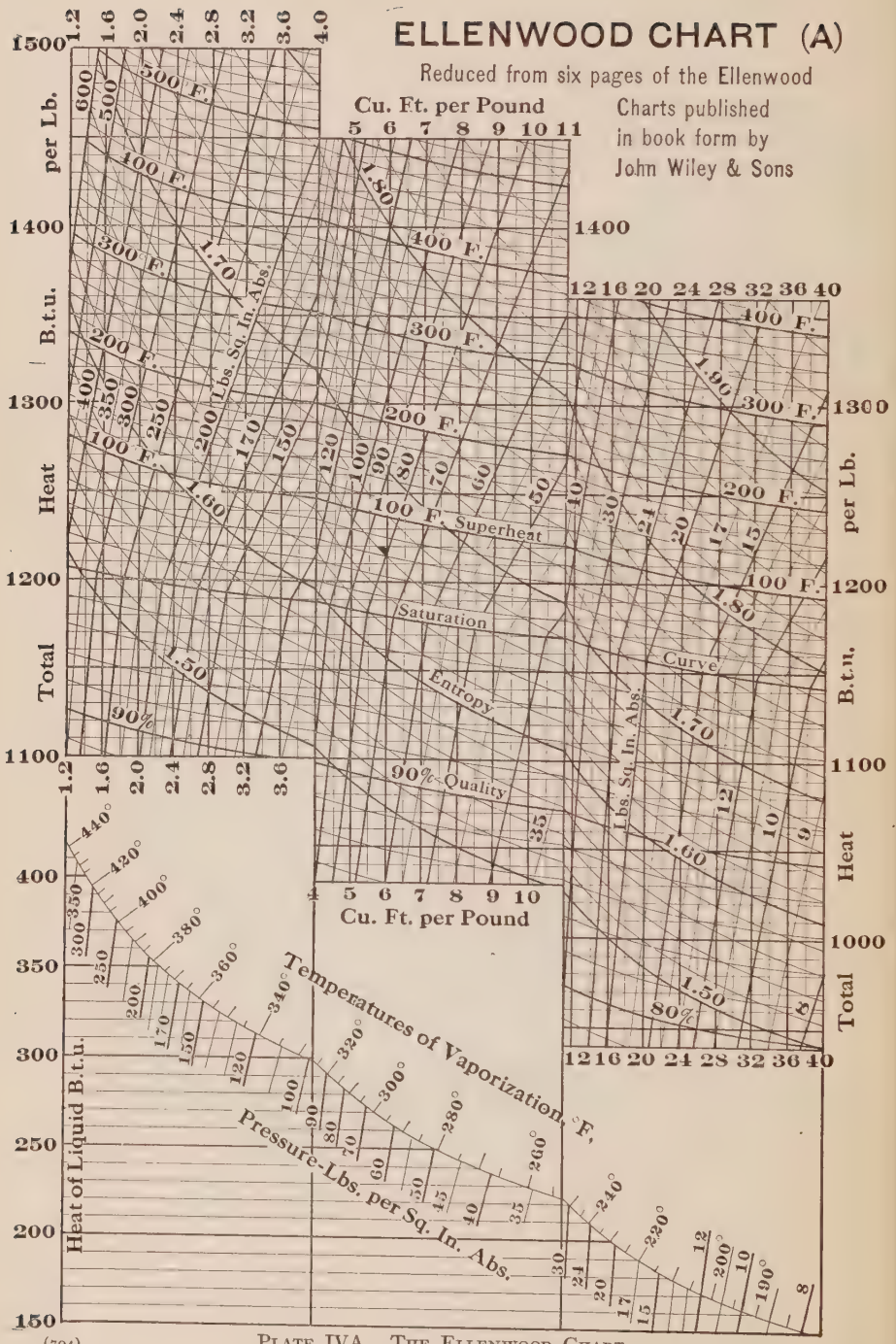


PLATE III. CONSTANT-PRESSURE EXTERNAL WORK CHART (ELLENWOOD).



(794). PLATE IVA. THE ELLENWOOD CHART.

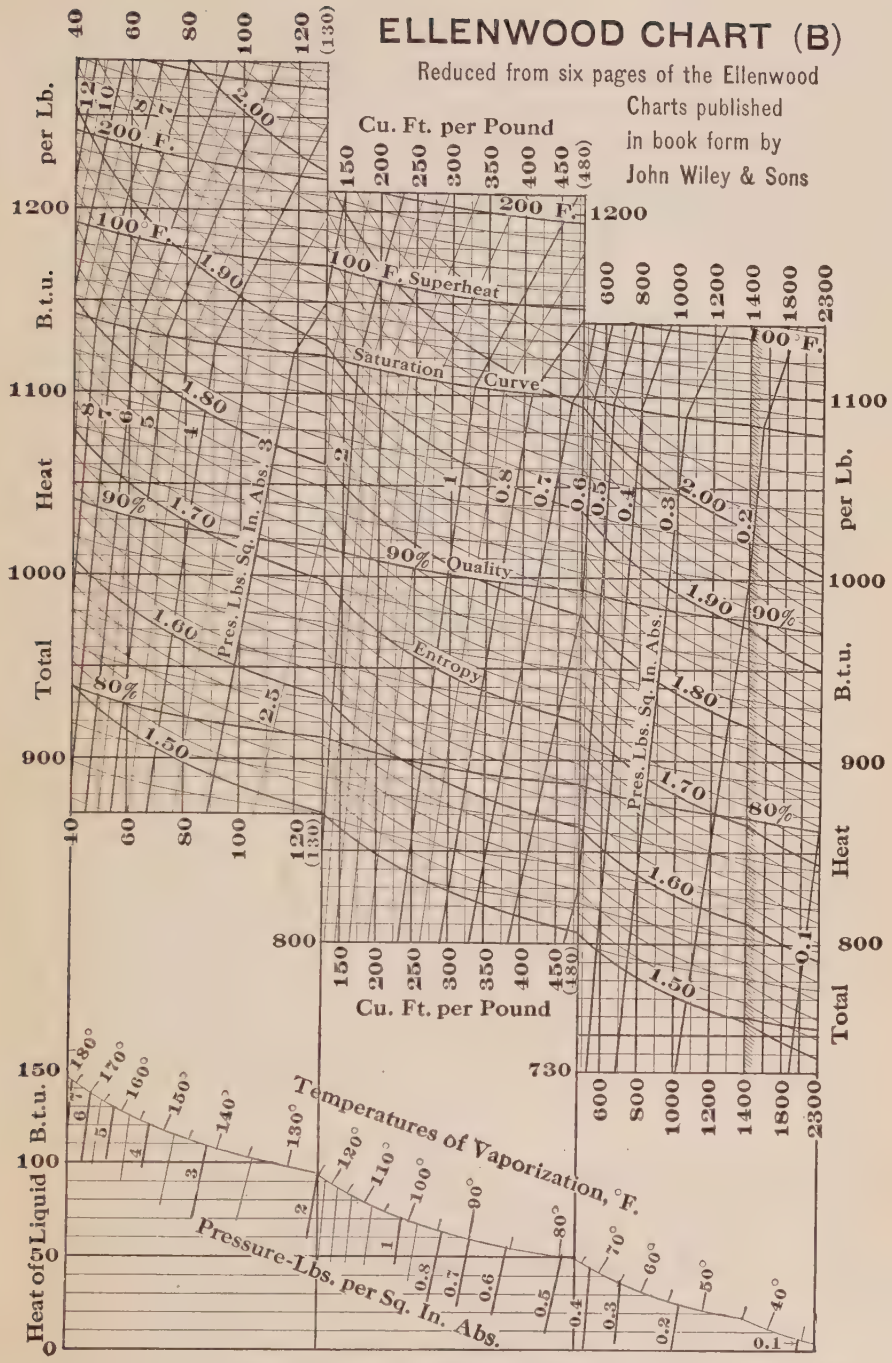
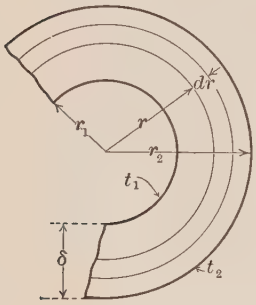

ELLENWOOD CHART (B)
Reduced from six pages of the Ellenwood Charts published in book form by John Wiley & Sons
Cu. Ft. per Pound
Total Heat B.t.u. per Lb.
Saturation Curve
Superheat
Quality
Entropy
Pres. Lbs. Sq. In. Abs.
Temperatures of Vaporization, °F.
Heat of Liquid B.t.u.
Pressure-Lbs. per Sq. In. Abs.

PLATE IVB. THE ELLENWOOD CHART (CONT.)

Heat Conduction through Cylindrical Walls. In the figure assume the temperature inside the cylinder to be t_1 and that at the outer surface to be t_2 (lower), the drop in temperature being θ. Then consider that there is a temperature drop of dt through the circular element of thickness dr and of radius equal to r.

Then if α is the specific heat of conductivity (from page 625)



$$dQ = 2\pi r \times \frac{\alpha\, dt}{dr},$$

$$\text{or} \quad \Delta Q = 2\pi\alpha \int_{r_2}^{r_1} \int_{t_2}^{t_1} \frac{r}{dr} dt$$

$$= 2\pi\alpha\,(t_1 - t_2) \log_e \frac{r_1}{r_2}$$

$$= 2\pi\alpha\theta \log_e \frac{r_1}{r_1 + \delta}$$

where δ is the thickness.

SYMBOLS.

(The numbers refer to the pages where the symbol is first used in a new sense.)

A, area; $A, \left(\frac{1}{778}\right)$; A, 502; Apu, 107; $ABEf$, 537; a (area); a_L, 328.

B (Baume); B_d. B_{dK}, B_I, B_{IK}, 364; $BCEf$, 537; B.P., 562; **B.t.u.**, 1; b.h.p., 186.

C, 257, 326; $\overline{C}$, 14; C_p, 35; C_v, 33; CEf, 188, 536; CEf_c, 640; CEf_h, 640; CF, 229; C.o.P., 735; c (const.).

D, 111, 579; DF, 325; d (differential coefficient); d, 130, 580; d.h.p., 186.

E, 582; Ef, 82; Ef_c, 164, 592, 641; Ef_h, 594, 641; Ef_n, 372, e (base for Naperian logs.); e.h.p., 186.

F, 36, 575; FE, 562; FEf, 536; f, 575; f.h.p, 186.

G, 662; GEf, 536; g (32.2 ft. sec.).

H, 338, 580; $H.P.$, 234; h, 257, 578; h.p., 180; h.p.-hr., 180.

I, 338; $I.P.$, 234; IEf, 189; i.h.p., 184.

J, 372.

K, 335, 572, 639, 698; K_p, 35; K_v, 34; KE, 372; k (const.).

L, 36, 184, 338, 575; $L.P.$, 234; $L.H.V.$, 488.

M, M', 502; MEf, 189; m, 495; m.e.p., 184.

n, 51, 184, 257, 643, 644, 646.

OEf, 190; OEf_d, OEf_k, 363.

P, 33; P_m, 324; Po, 698; p, 325; p_m, 184, 325; p_{mH}, p_{mL}, p_{mR}, 333; p_R, 334.

Q, 8; q, 105; q_f, q_f', 651.

R, 32, 258, 315, 328, 372, 626; REf, 188; r, 49, 108, 258, 325; r_H, r_L, r_T, 331.

S, 572, 626.

T, 30; T_s, 111; T_v, 105; $TBEf$, 537; $TDEf$, 190; $TDEf_K$, 365; $TIEf$, 190; T.U.E., 561; t, 30, 638; t_a 499; t_c, t_d, 674; t_f, 499, 653; t_f', 653; t_i, t_m, 672; t_v, 105.

U.E., 561; u, 107, 133, 140, 371.

V, 32, 286, 462; $\mathbf{V}$, 33, 133; V_e, 680; V_{es}, 719; $\mathbf{V}_s$, 137; V_w, 680; VEf, 720; v, 286, 371.

W, 8, 14, 197, 258; W_c, 239; W_d, 191, 363; W_{dK}, 363; W_i, 191; W_I, W_{IK}, 363; W_j, 239; w, 206, 372, 496, 653, 672; w', 654; w_c, 638; w_f, 205; w_f', 228; w_k, 205; w_h, 638; w_H, 496; w_N, 496.

X, 481; x, 110, 479; x_f, x_k, 220.

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Z, 647; z, 575;

α, 277, 625; β, 627; γ, 38.

Δ (finite change); ΔE, 11; ΔE_p, ΔE_s, 158; ΔE_{sx}, 159; ΔI, ΔQ, 11; ΔQ_D, ΔQ_s, 111; ΔQ_{xp}, 110; ΔQ_{xs}, 111; ΔQ_{yp}, 109; ΔS, 11; $\Delta\phi$, 66; $\Delta\phi_D$, $\Delta\phi_l$, 119; $\Delta\phi_s$, 120; $\Delta\phi_{sa}$, 120; $\Delta\phi_v$, 119; $\Delta\phi_x$, 120.

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λ, 108; ρ, 108; ϕ, 65; ω, 258.

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